VALVE GEARS

American School

Chicago, U. S. A.

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Prepared especially for the instruction and training of students of the American School

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VALVE CHARACTERISTICS

Function. Steam enters the cylinder of a steam engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose, one or more valves are moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, the economy of the engine may be seriously impaired by improper valve setting. The design and adjustment of these valves play a very important part in the efficient action of the steam engine.

A valve gear is a mechanism consisting of a combination of slotted links, eccentrics, rods, levers, and other devices, designed to operate valves of various types. The valve gear is separate and distinct from the valve. It operates the valve or valves but, strictly speaking, is not a part of them. This being true, one type of valve gear may be applied or used in connection with several different types of valves. For instance, the Stephenson gear may be used to operate a plain slide valve on one engine, a piston valve having either inside or outside admission on another, while a third may be attached to a more complicated form of valve mechanism. It should be borne in mind, therefore, that the valve gear is a separate and distinct part of the steam engine and that its function is to impart motion to the valve or valves.

The valves, in turn, perform the following functions during the engine cycle:

(1) *Admission.* This begins when the valve opens to admit steam to the cylinder.

(2) *Cut-Off.* This is the point at which the valve closes to cut off the admission of steam.

(3) *Expansion.* This takes place from cut-off to release.

(4) *Release.* This begins when the exhaust port is opened.

(5) *Compression.* This begins when the exhaust port is closed.

There may be a single valve to regulate admission and exhaust or there may be a double set of valves, one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved either by a rod or by some device that releases at the proper time, allowing the port to close suddenly under the influence of counterweights, springs, or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second class belong such valves as the Corliss, the Brown, and others.

The simplest type of valve is the plain slide, or *D*, valve as shown in Fig. 1, in which *V* is the valve, *R* is the valve rod, *K* is the exhaust cavity, P_1 and P_2 are the steam ports, *E* is the exhaust port, *A B* is the valve seat, and *D M* are the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that steam pressure on the valve will make a steam-tight fit and cause as little friction as possible when the valve moves back and forth. Furthermore, the length of the seat *A B* must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke to slightly overtravel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

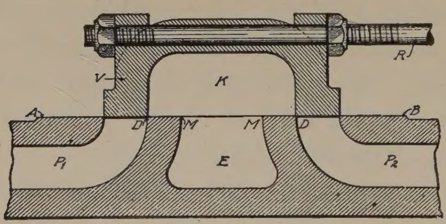

Fig. 1. Plain Slide, or D, valve

Eccentric. The valve usually receives its motion from an eccentric, which is simply a disk keyed to the shaft in such a manner that the center of the disk and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disk moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equiva-

lent to the action of a crank whose length is equal to the distance between the center of the eccentric and that of the shaft

Fig. 2 represents the essentials of an ordinary eccentric. O_1 is the center of the shaft, O_2 is the center of the eccentric disk E, and S is a collar encircling the eccentric and attached to the valve rod R. As the eccentric turns in the strap, the point O_2 moves in the dotted circle around O_1 and the point A_1 also moves in a circle. When half a revolution is accomplished, the point O_2 will be at O_3, the point A_1 will be at A_2, and the eccentric strap and valve rod will be in the position indicated by the dotted lines. The distance O_1O_2 of the

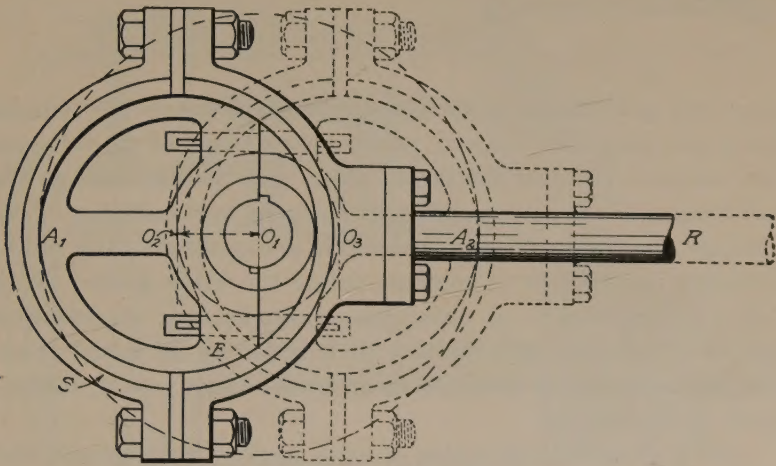

Fig. 2. Details of Ordinary Eccentric

center of the eccentric from the center of the shaft is known as the *eccentricity*, or *throw*, of the eccentric. The *travel* of the valve is twice the eccentricity.

Since the eccentric transmits the motion of the revolving shaft to the valve, it will be necessary to study the relative motions of crank and eccentric in order to obtain a clear idea of the steam distribution. This relation will be developed in connection with the discussion of the valve action which follows.

Valve Motion. *Valve without Lap.* Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide valve placed in mid-position* and so constructed that

* A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

in this position it just covers the steam ports. Referring to Fig. 1, which shows the same type of valve drawn to a larger scale, suppose the valve is moved a slight distance to the right; the port P_1 is then

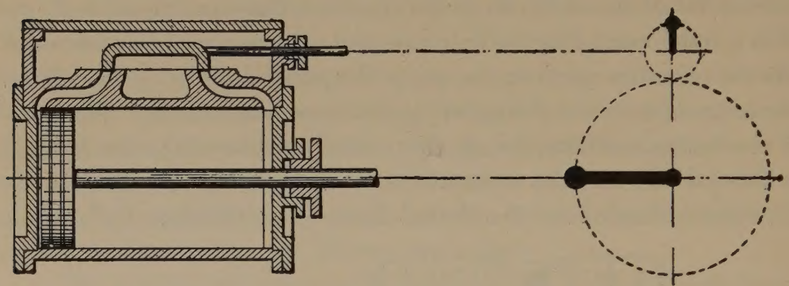

Fig. 3. Cylinder Details_Showing Plain Slide Valve without Lap in Mid-Position

uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P_1 opens to live steam, the port P_2 opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust to continue during the whole stroke. With such a valve, there is no expansion or compression; the indicator card is a rectangle, and the m. e. p. is equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center

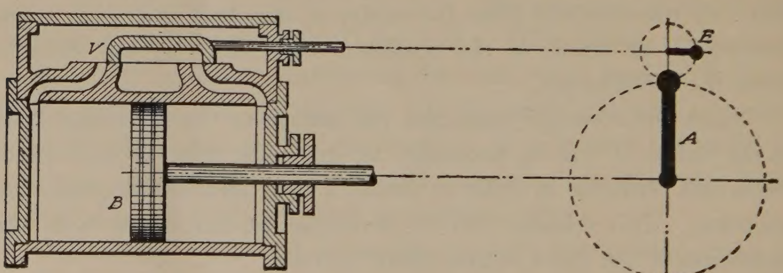

Fig. 4. Position of Piston and Valve in Cylinder Shown in Fig. 3, after One-Half Stroke

line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with

the length of the eccentric rod that the angularity of the eccentric rod is very much less than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of the crank *A*, piston *B*, eccentric *E*, and valve *V*, when the crank has made a quarter turn or the piston has moved to about half-stroke. The eccentric is now in its extreme position to the right, the valve has its maximum dis-

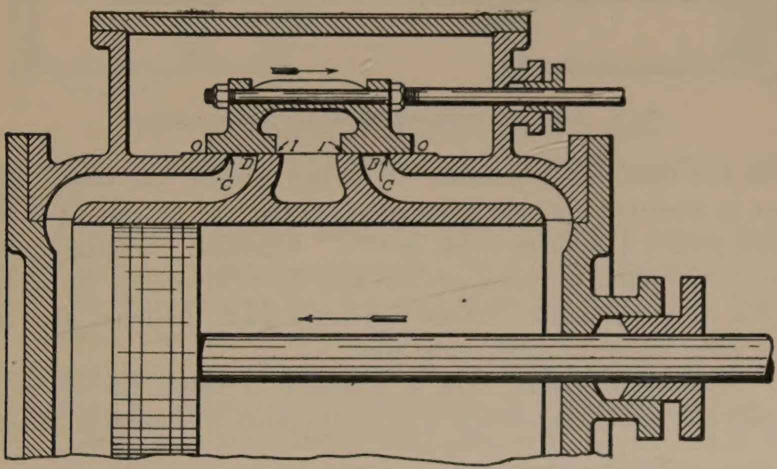

Fig. 5. Details of Cylinder, Showing Valve with Lap

placement, and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small engines and, since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that

the valve overlaps the steam ports when in mid-position is called the *lap* of the valve. In Fig. 5, *D I* is the *inside lap* and *O C* is the *outside lap.*

It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expansion and compression possible. It is also evident that steam can not be admitted until the valve uncovers the port by moving from mid-position a distance equal to *O C.* Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the valve begins to uncover the port.

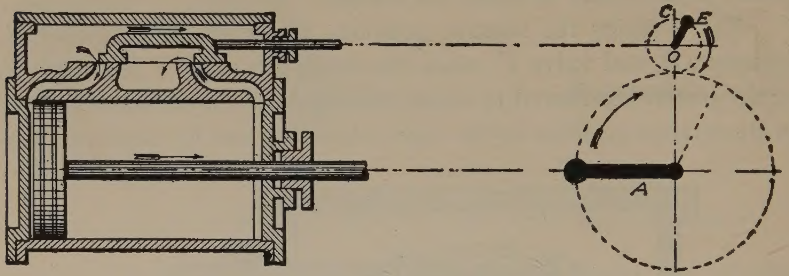

Fig. 6. Valve with Inside and Outside Lap Set for Admission

Analysis of Motion. Fig. 6 represents a valve, having both inside and outside lap, which is set at the point of admission. Since the valve must move over a distance equal to the outside lap in order that admission may take place under proper conditions, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be in advance of the

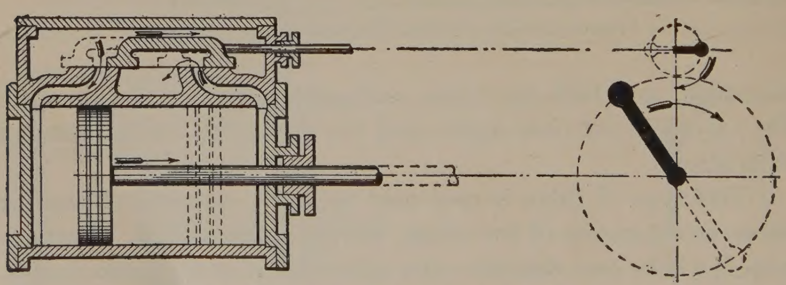

Fig. 7. Valve Set at Maximum Displacement

right-angle point by an amount equal to the angle EOC, known as the *angular advance.*

The maximum displacement of the valve is attained when the eccentric is horizontal, as shown in Fig. 7. In this position, both the steam and the exhaust ports are wide open, and any further

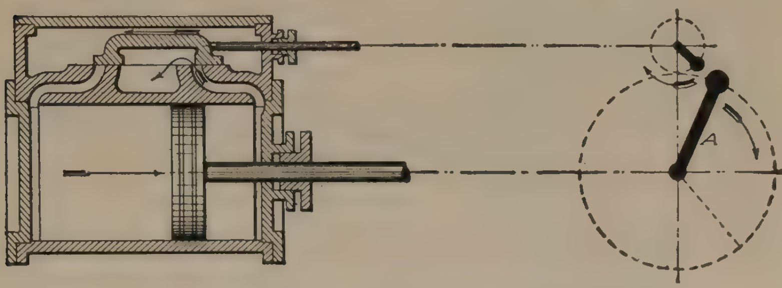

Fig. 8. Valve Position with Steam Port Closed on Head End

motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10. The dotted lines of Fig. 7 show the valve in its extreme position to the left, while the

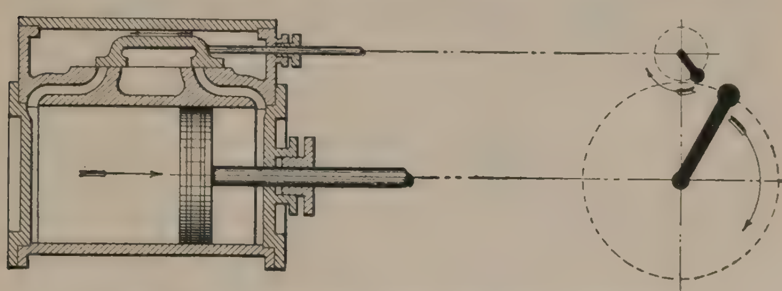

Fig. 9. Valve Position with Exhaust Port Closed on Crank End

dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end. While the crank A is moving from the position shown in Fig. 6 to that in Fig. 8, steam is being admitted to the head end and being exhausted from

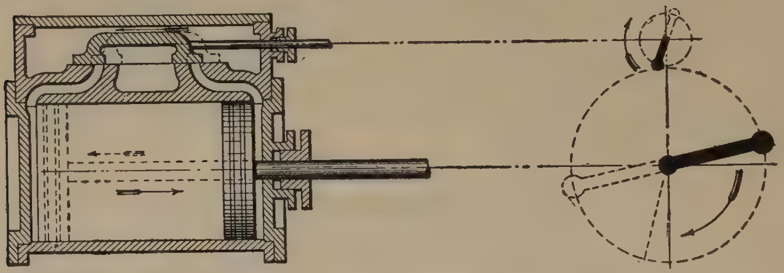

Fig. 10. Position of Valve and Cylinder for Head-End Release

the crank end. As the inside lap is less than the outside lap, the exhaust continues longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric, and valve when the exhaust closes on the crank end and compression begins. Between these two positions, the steam is expanding in the head end and exhausting from the crank end. Between the positions of Figs. 9 and 10, both ports are entirely closed, and expansion is taking place in the head end and compression in the crank end. In Fig. 10 is shown the position of the valve for head-end

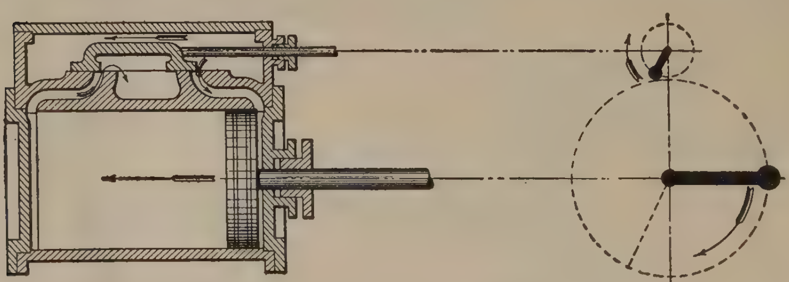

Fig. 11. Position of Valve and Cylinder for Crank-End Admission

release. Fig. 11 shows admission at the crank end of the cylinder and marks the end of crank-end compression.

Effect of Change of Lap. By referring to Figs. 6 to 11, the effect of any change of lap may at once be observed. If the outside lap is increased, the valve must move farther from mid-position before

admission will occur and on the return, after the maximum displacement is reached, the greater outside lap will close the port sooner, and the point of cut-off shown in Fig. 8 will be reached before the crank reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

On the other hand, if the inside lap is increased, the valve must move farther before release occurs and the crank angle will be greater than that shown in Fig. 10, while on the return to the dotted position, the port will close earlier and make an earlier compression. The crank angle will be less than is there shown. Decreasing the inside lap will cause earlier release and later compression.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap* and the inside lap is called the *exhaust lap*.

Lead. If a valve having lap is in mid-position, the port is closed and the engine can not start, because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke, it is necessary that the eccentric be set more than 90 degrees ahead of the crank as already mentioned, thus making the eccentric radius take an angular advance EOC, as shown in Fig. 6. In order that the ports and all clearance space may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke, the valve will have a position as shown in Fig. 12, the port being open the distance AB, the lead of the valve. This causes the eccentric to be moved forward a slight amount in excess of the lap angle. This excess is called the *angle of lead*.

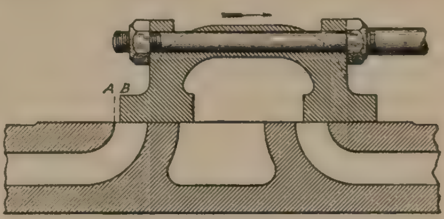

Fig. 12. Position of Valve Showing Lead

In Fig. 13, O_2R_2 represents the position of the crank at the beginning of the stroke, LO_1A_1 the lap angle, and $A_1O_1A_2$ the angle of lead. The eccentric, to give lead, must be set at the angle $R_1O_1A_2$ ahead of the crank or 90 degrees plus the angular advance. In large

high-speed engines, a liberal lead is essential in order that the ports and clearance space may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full steam pressure will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line, as shown in Fig. 14. Too much lead, on the other hand, will cause too early an admission, as shown in Fig. 15.

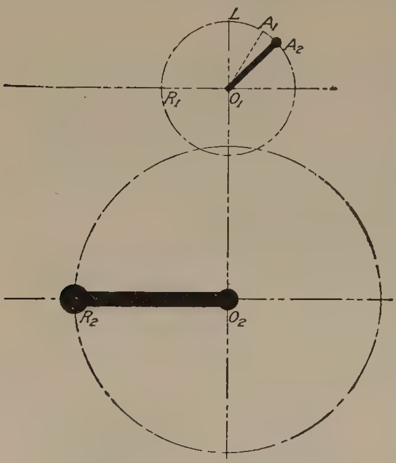

Fig. 13. Diagram Showing Lap Angle and Angle of Lead

If the angular advance is increased, the eccentric will be moved further ahead of the crank, and consequently it will arrive at each of the events sooner than before. If, then, the angular advance is increased, all of the events of the stroke will occur earlier.

Effect of Lead. From the foregoing discussion of lead, it is evident that its effect is to permit steam to enter the cylinder before the end of the stroke, which tends to provide an abundance of steam behind the piston when starting the return stroke and throughout the period of admission. It also promotes smooth running of the engine by furnishing a cushion or retarding force to the moving parts, thereby eliminating the "knocks" or "pounds" incident to lost motion. Since the effect of lost motion depends upon the weight

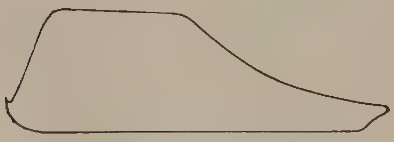

Fig. 14. Indicator Diagram Showing Effect of No Lead

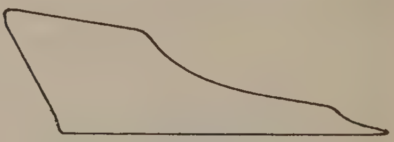

Fig. 15. Indicator Diagram Showing Too Early Admission

and velocity of the reciprocating parts, it is evident that the amount of lead required will vary for different engines and for the same engine running at different speeds. The exact amount of lead can not be determined except by trial and by use of the steam engine indicator.

When experimenting for the determination of the proper amount of lead for a specific case, it will be necessary to gradually increase the angular advance until smooth running is obtained. After this result is obtained, indicator cards should be taken to see if the lead is excessive, in which case the valve must be adjusted until the desired conditions are obtained. Since lead permits steam to act against the piston before the end of the stroke, it results in negative work, hence the amount of lead should not be excessive. An amount of lead sufficient to insure the filling of the clearance space is permissible, but very much more than this is detrimental to the economic performance of the engine.

Analytical Summary of Valve Terms. Thus far in discussing the plain slide valve, a number of terms have been used that are of primary importance and must be thoroughly understood in order to properly grasp much that is yet to be studied. It seems advisable, therefore, that a recapitulation of the terms used be presented.

Mid-Position. A valve is said to be in mid-position when the center of the valve and valve seat coincide. When in this position, the steam ports are all closed.

Displacement. The displacement or a valve is the amount the valve has been moved either to the right or left of its mid-position. In Fig. 4, the valve has moved to the right a distance equal to the width of the steam port, hence in this instance the displacement of the valve is equal to the width of the steam port.

Valve Travel. The travel of the valve is the distance the valve travels in moving from one extreme position to the other. The travel of the valve is twice the eccentricity, or throw of the eccentric.

Eccentricity. The eccentricity, or throw of the eccentric, is the distance between the center of the shaft and the center of the eccentric. It is equivalent to a crank, the length of which is one-half the valve travel. For instance, if the valve travel of an engine is 6 inches, the eccentricity, or throw of the eccentric, would be 3 inches, or one-half of the valve travel.

Lap. The amount that the valve extends over the steam port when in mid-position is called steam lap or often spoken of as the lap of the valve. The steam lap is equal to $O C$ in Fig. 5. In Fig. 5, it is obvious that when the valve is in mid-position, the distance $D I$ is called exhaust lap. Steam lap and exhaust lap are frequently

spoken of as outside and inside lap, respectively. The effect of the exhaust lap is to delay exhaust and hasten compression.

Very frequently a valve does not have any exhaust lap and there is a small port opening between the cylinder and the exhaust cavity when the valve is in mid-position, as shown at *A*, Fig. 19. In such a case, the valve is said to have *inside clearance*. The effect of inside clearance is opposite to that of exhaust lap, namely, it delays compression and hastens exhaust, and insures a minimum amount of back pressure.

Lead. By the term *lead* is meant the amount the steam port is open when the engine is on either dead center.

Angle of Advance. It was noted in Fig. 1 that the crank and eccentric were exactly 90 degrees apart and that admission occurred at the beginning and cut-off at the end of the stroke. On account of economic reasons, this is not a good arrangement. Hence we find that the valves on all engines have lap and are set to give the necessary amount of lead. In order to obtain lead when the engine is on dead center with a valve having lap, it is necessary to turn the eccentric ahead, in the direction the engine is to run, through such an angle that the valve will be displaced by an amount equal to the lap plus the lead. The angle measuring this displacement is the sum of the angle of lap and the angle of lead. If there is no lead, this angle would be decreased by the angle of lead. The sum of the angle of lap and the angle of lead is frequently designated as the angle of advance. The angularity between the eccentric and the crank then becomes equal to 90 degrees, plus or minus the angle of advance according to the type of valve and gear.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston. If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod varies from four to eight times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine when the engine is off dead center, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

To Find Displacement of Valve. The circle, Fig. 16, represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center line of the engine. When the eccentric is at OL, the valve is in mid-position. At OR the valve has moved from mid-position an amount ON, found by dropping a perpendicular from R to the center line XY. If the angularity of the connecting rod could be neglected, the piston displacement could be found in the same manner.

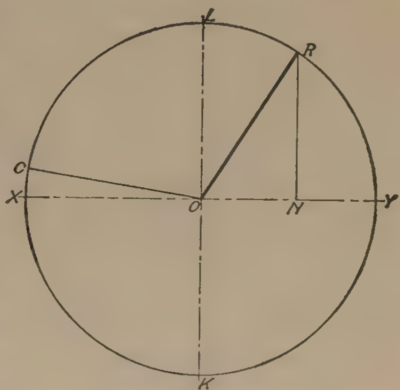

Fig. 16. Eccentric Circle Showing Relative Positions of Crank and Eccentric

To Find Displacement of Piston. To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure, AB represents the cylinder, P_1 the piston, H_1 the crosshead, H_1R the connecting rod, and OR the crank. Suppose now the engine should stop and the piston be clamped in this position. The piston displacement would be represented by AP_1. If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a

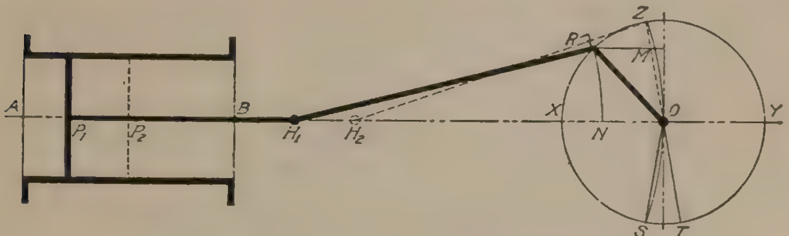

Fig. 17. Diagram for Finding Displacement of Piston

horizontal position, the point R would describe the arc of a circle RN, and XN would represent the piston displacement and would be equal to AP_1. Suppose now that in this disconnected way, the

piston, crosshead, and connecting rod were moved forward until the end of the rod came to O. P_1 would then be at P_2 and the piston would be in the middle of its stroke. Now suppose the end of the rod were swung up to its proper position on the crank-pin circle, the piston remaining stationary. The end of the rod would describe an arc OZ; the crank pin would be at Z, less than a quarter revolution from X; while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end, and that XOZ represented the corresponding crank angle. On the return stroke, the valve would cut off at the same crank angle YOT, which is equal to XOZ, and OT would represent the crank position for cut-off on the return, or crank-end, stroke. The piston, as we have just seen, will not be at half stroke except

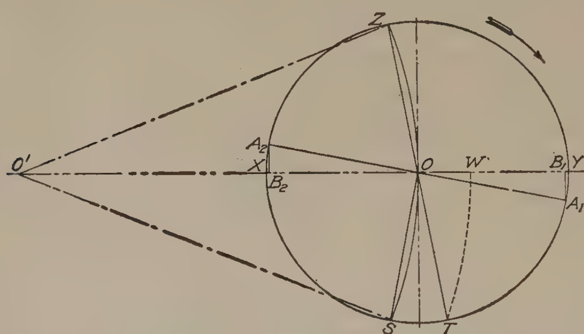

Fig. 18. Crank and Eccentric Diagram for Engine Shown in Fig. 17

when the crank is at OZ or OS. Consequently, the crank position OT is less than half stroke and cut-off occurs earlier at crank end than at head end. When the crank is at OZ, the eccentric will be at OA_1, Fig. 18, assuming the valve to have no lap, and the valve displacement will be OB_1. When the crank is at OT, the eccentric will be at OA_2 and the valve displacement will be OB_2, which is equal to OB_1, the displacement of the valve at cut-off on the head end. The piston displacement will be OX on the head end and WY on the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-Off. It has been pointed out that lengthening the outside lap makes the cut-off earlier, and that shortening

the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

But altering the outside lap changes the lead, as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if the proper care is not taken.

Unequal lead is of less consequence on a low-speed engine than on a high-speed engine. On low-speed engines, the cut-off may be

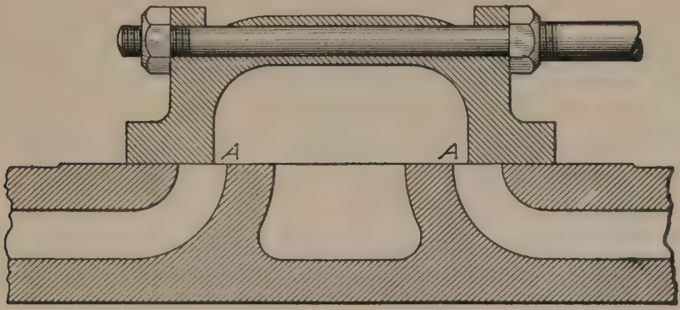

Fig. 19. Valve in Mid-Position Showing Inside Clearence, or Negative Lap

equalized at the expense of lead with beneficial results, but on high-speed engines, this is not true. A high-speed engine requires more lead than a low-speed engine, for there is relatively less time in each stroke for the clearance space to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position. The amount of this opening is called *inside clearance*, or *negative lap*. This is illustrated at *A*, Fig. 19.

Rocker. Sometimes it happens that the valve stem and eccentric rod can not be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases, a rocker may be used.

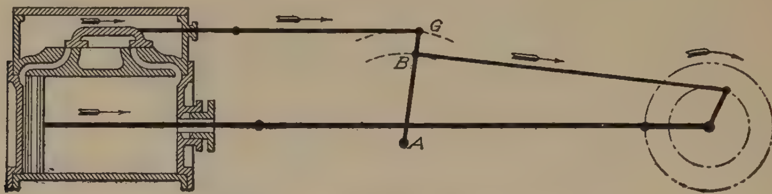

Fig. 20. Valve with Rocker Arrangement

Fig. 20 shows a valve that is not in line with the eccentric. An instance where this occurs is in horizontal engines when the valve is set on top of the cylinder instead of on one side. By means of the rocker *A G*, the valve may receive its proper motion.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as is shown in Fig. 21 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions and in order to give the valve the same motion as in Fig. 20, the eccentric must be moved 180 degrees ahead of the position there shown.

If *A B* is less than *A G*, the valve travel will be greater than twice the eccentricity, in proportion as *A G* is greater than *A B*. In all

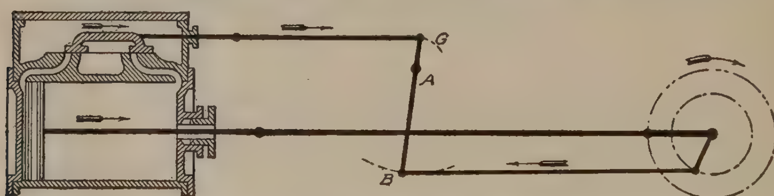

Fig. 21. Arrangement of Rocker by which Valve Stem and Eccentric Rod Move in Opposite Directions

cases, the valve travel is in the same proportion to twice the eccentricity as *A G* is to *A B*. Thus, if the valve travel is $4\frac{1}{2}$ inches, *A B* is 15 inches, and *A G* is 18 inches, then $\frac{15}{18}\times4\frac{1}{2}=3\frac{3}{4}$ inches, will equal twice the eccentricity.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by

a proper proportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner, the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS

Zeuner Diagrams. In order to study the movements of the valves, the effects of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire

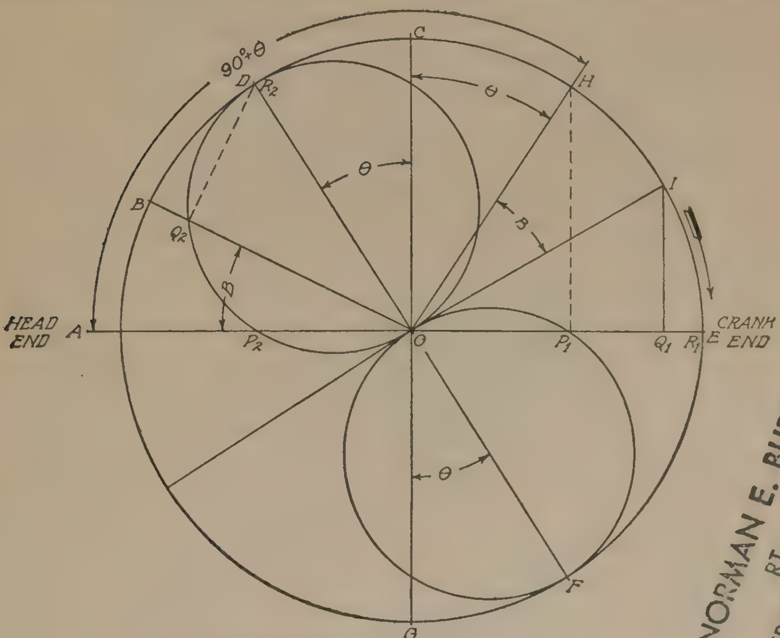

Fig. 22. Zeuner Diagram for Valve Analysis

a knowledge of the valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner and, to avoid complexity, we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

The function of the Zeuner diagram is to show the relation between the valve positions and crank positions. This relation being known, it is a simple matter to obtain the eccentric and piston positions.

In Fig. 22, $A\,O\,E$ represents the valve travel, and the center of the eccentric will move in the circle $A\,C\,E\,G$. It is assumed, also, that this circle represents the path of the crank center, hence this circle will be known as the *crank and valve circle.* OA is the position of the crank and $O\,H$ is the corresponding position of the eccentric, when the engine is on the head-end dead center. Since this valve has lap, and since admission must occur before the end of the stroke, it is evident that the eccentric must precede the crank by 90 degrees plus the angle of advance θ. From H drop a perpendicular line upon the center line $A\,O\,E$, thus locating the point P_1. The distance OP_1 is the amount the valve has been moved to the right of its mid-position when the crank is on dead center. Since the diagram gives the relation between crank and valve positions, the displacement of the valve $O\,P_1$ can be laid off from O on the crank position OA, thus establishing the point P_2. Turn the crank through an angle B to the position $O\,B$. The eccentric will move through the same angle and will be found at I. Draw the perpendicular line IQ_1, and OQ_1 represents the displacement of the valve for the crank position $O\,B$. Lay off $O\,Q_1$ on $O\,B$, establishing the point Q_2. Continue the rotation of the crank until the point D is reached. The eccentric then will be found at E, and the valve will have its greatest displacement $O\,R_1$ to the right of its mid-position. It is evident that $O\,R_1$ is equal to $O\,D$. If the rotation of the crank be continued in the direction of the arrow, the valve will return from its extreme position on the right and will approach its mid-position. By locating on the various crank positions the corresponding valve displacement, a series of points as P_2, Q_2, R_2, etc., will be obtained, all of which will lie on the circumference of a circle, as $OP_2Q_2R_2$, the diameter $O\,D$ of which will make an angle θ equal to the angle of advance laid off to the left of the vertical $O\,C$. If this operaation be continued for a complete revolution, a series of points will be established in the lower quadrant, establishing a circle OP_1F, the diameter of which will be a continuation of $O\,D$ and, therefore, will make an angle θ with the vertical but will lie on the right of

the vertical line COG. These two circles are called *valve circles*, and they represent the movement of the valve to the right and left of its mid-position and, as previously stated, represent the amount the valve has moved for any crank position such as OB.

Having established the valve circles, it is a simple matter to obtain the valve displacement for the position OB, which, in this case, would be the distance OQ_2 cut off from OB by the valve circle. It can be proven that OQ_2 is the valve displacement by comparing the two right triangles OIQ_1 and ODQ_2. They are equal because

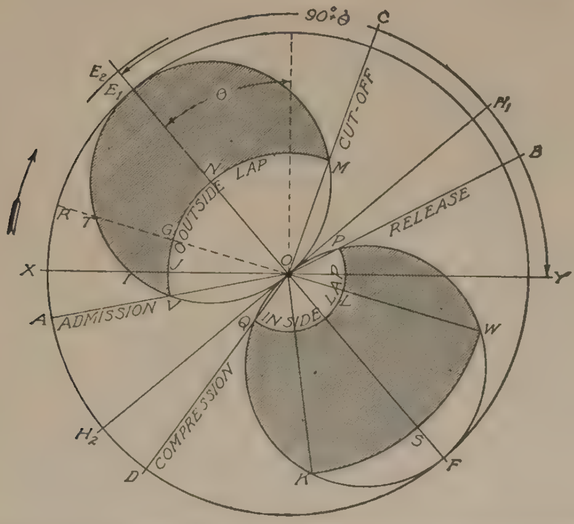

Fig. 23. Diagram Showing Study of Valve Motion for Head End Only

they are similar and have two corresponding sides OD and OI equal. Therefore, OQ_2 equals OQ_1. This being true for any crank position, it is true for all crank positions.

Study of Valve Motion from Diagram. Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. In Fig. 23 let XY represent the valve travel; then the circle XE_1YF will represent the path of the center of the eccentric. Let θ be the angle of advance and lay off E_1O toward the crank, making an angle θ with the vertical. Produce E_1O to F, and on OE_1 and OF as diameters draw the valve circles as shown. With O as a center and OV, equal the outside lap, as a

radius, draw an arc intersecting the upper valve circle at V and M. Lay off $O\,P$ equal to the inside lap and with O as a center and $O\,P$ as a radius, draw an arc intersecting the valve circle at P and Q. Draw the crank-position line $A\,O$ passing through V. Then, when the crank is in this position, the displacement of the valve is equal to $O\,V$ (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the *lead angle*. The valve has lead and, therefore, the admission takes place before the end of the stroke. When the crank reaches the position OE_1, the displacement of the valve is equal to the eccentricity OE_1, and is at a maximum. Further motion of the piston causes the valve to move toward mid-position until, at the crank position $O\,C$, the displacement $O\,M$ is again equal to the outside lap and the valve has reached the point of cut-off. When the position $O\,H_1$ is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point, the valve is in mid-position.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position $O\,B$, the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at $O\,D$ its displacement from mid-position is again equal to $O\,Q$, equals OP the inside lap, and compression takes place. At $O\,H_2$ the valve is again in mid-position. At OX the displacement of the valve is OI, but since the valve has to move the distance OJ before the port begins to open, IJ must represent the port opening when the crank is on dead center, and by definition we know that lead is the amount of port opening at this position. Therefore, $I\,J$ represents the lead.

At the position R, the port is open an amount equal to $T\,G$; at E_1 the opening is a maximum equal to $E_1\,N$; at C the port is on the point of closing and there is no opening. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to $O\,W$ and it will remain wide open while the crank swings from $O\,W$ to $O\,K$.

If the width of steam port in addition to the outside lap were laid off on the other valve circle, it would fall at E_2. For the admission port to be wide open, the displacement of the valve would have

to be equal to OE_2 which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to E_1E_2 when the valve has reached its maximum displacement.

Fig. 23, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

Properties of Zeuner Diagrams. The Zeuner diagram deals with admission, cut-off, release, compression, lead, valve travel, angle of

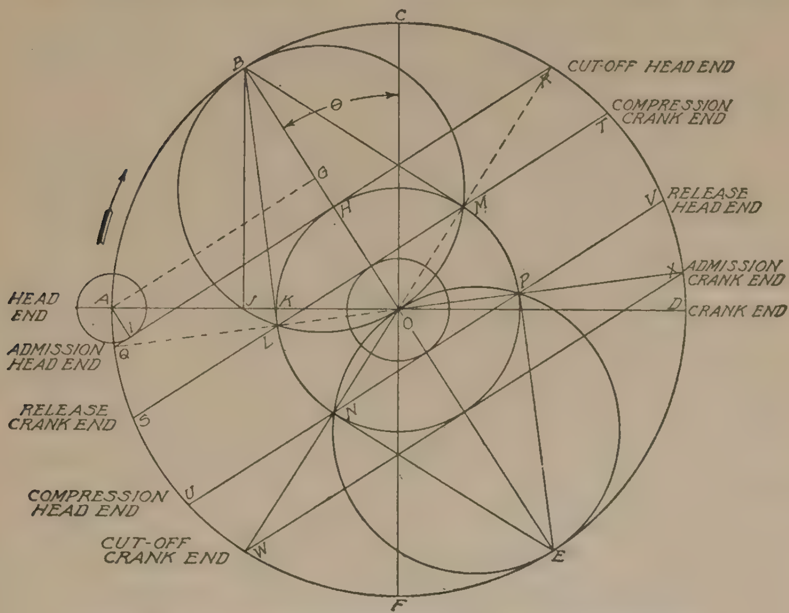

Fig. 24. Diagram Analysis for Movement of a Direct Valve as Regards Head End of Cylinder

advance, maximum and minimum port opening, steam lap, and exhaust lap. Generally, if four of these be given, the others can be found. It is evident, therefore, that there are a great many possible combinations, hence it is necessary to have definitely in mind and clearly understood the properties of the Zeuner diagram. The proofs given are for the movement of a direct valve as regards the head end of the cylinder. All letters refer to Fig. 24.

(1) The figure is symmetrical on the line BE. In the semicircles OLB and OMB, OL equals OM, each being the radius of the steam-lap circle. Since OL equals OM, the arcs which they subtend are equal, therefore, the arcs LJB and MB are equal. This makes the angles LOB and MOB equal because they are measured by equal arcs. Therefore, BO bisects the angle LOM, and in a similar way it can be proved that OE bisects the angle NOP.

(2) The line BM is perpendicular to OMR and is tangent to the steam-lap circle. The angle BMO is a right angle because it is inscribed in a semicircle. Therefore, BM is tangent to the steam-lap circle and is perpendicular to the crank position OMR.

(3) The line joining the admission and cut-off points for the head end is perpendicular to BO and is tangent to the steam-lap circle.

The triangle QOR is an isosceles triangle and, as demonstrated above, BO bisects the angle QOR, hence BO is perpendicular to the base QR. To prove that QR is tangent to the steam-lap circle, it is necessary to show that the distance OH measured on BO is equal to OM, the radius of the steam-lap circle. The right triangles BOM and HOR are equal, having two sides equal and one common angle. Hence, OH is equal to OM.

(4) The line BJ is perpendicular to AO. The angle BJO is a right angle, being inscribed in a semicircle.

(5) The radius of the circle AI with center at A and tangent to QR, is equal to the lead JK.

From the center A draw AG parallel to IH. In the right triangles BJO and AGO, the angle AGO equals BJO, being right angles. BO equals AO. The angle AOH is common to both triangles, therefore, they are equal. Hence, OJ equals OG. But OK equals OH. Therefore, GH equals JK, equals AI, which is the lead.

By using Fig. 24 at all times as a reference figure and bearing in mind the things it tells, no great difficulty should be encountered in solving problems. To illustrate the principles set forth above and to give an idea of the practical use of the Zeuner diagram, several problems will be worked out as an indication of what may be done.

ILLUSTRATIVE PROBLEMS

In designing a slide valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point, and the engine must have a certain lead. Then, with the aid of a Zeuner diagram, the remaining proportions of the valve may be determined.

Example 1. Given a valve travel of 3 inches, exhaust lap of $\frac{3}{4}$ inch, angular advance of 35 degrees, and crank angle at cut-off of 115 degrees. Determine the laps, the lead, and the crank angles at admission, compression, and release.

Solution. In Fig. 25, let $X\ Y$ represent the valve travel of 3 inches. Draw $O\ M$ perpendicular to $X\ Y$, and on $X\ Y$ as a diameter draw the circle

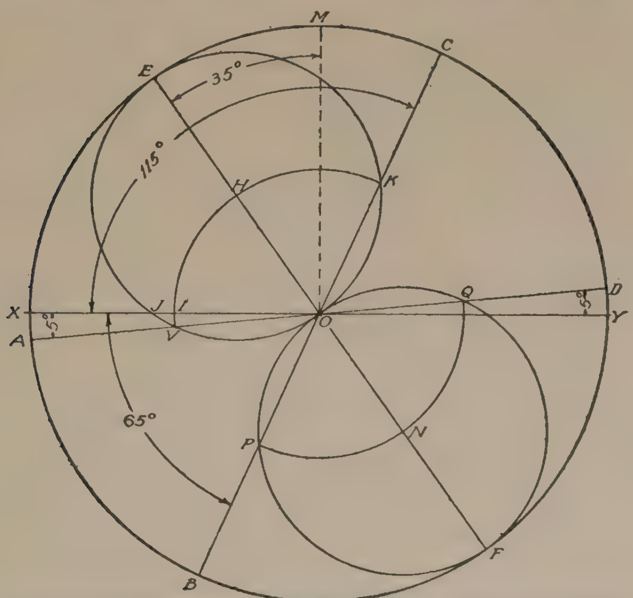

Fig. 25. Zeuner Diagram for Finding Laps, Lead, and Crank Angles

$X\ M\ Y\ F$ representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle $M\ O\ E$ to represent the angular advance of 35 degrees so that the angle $X\ O\ E$ is equal to 90 degrees minus the angular advance. Produce $E\ O$ to F. Then on $O\ E$ and $O\ F$ as diameters, draw the valve circles. The eccentricity $O\ E$ or $O\ F$, if no rocker is used, will be half the valve travel. Lay off the angle $X\ O\ C$ to represent the crank angle at cut-off of 115 degrees, and $O\ K$ will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc $K\ I$, known as the lap circle, and it will cut the valve circle again at V. When the valve is again the distance $O\ V$, the out-

side lap from mid-position, admission will take place. Draw the line $O\ V\ A$ and this will represent the position of the crank at admission.

When the crank is at $O\ X$, the valve displacement is equal to $O\ J$. This is at dead center, and the valve is open the amount $I\ J$, for it has moved this distance more than the outside lap. Therefore, $I\ J$ is the lead for this end.

Now on the other valve circle, draw the arc $P\ Q$ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to $O\ Q$, the exhaust port has just commenced to open, and the engine is at release. In the same way, when the valve displacement is equal to $O\ P$, the port begins to close and the engine is at compression. $O\ Q\ D$ represents the crank position at release and $O\ P\ B$ the crank position at compression.

The results are tabulated as follows:

Outside lap $O\ K$	$=\frac{3}{4}$ inch
Angle of lead $X\ O\ A$	$=5$ degrees
Linear lead $I\ J$	$=\frac{3}{32}$ inch
Max. port opening for admission $H\ E$	$=\frac{3}{4}$ inch
Crank angle at release $X\ O\ D$	$=185$ degrees
Crank angle at compression $X\ O\ B$	$=$ 65 degrees
Max. port opening for exhaust $F\ N$	$=\frac{3}{4}$ inch

Fig. 25 is drawn full size, and all of the above measurements may readily be verified. This figure is drawn for the head end only. If the crank angle at cut-off is the same on both ends, the Zeuner diagram for the crank end will be exactly like Fig. 25.

Example 2. Given a lead $\frac{1}{16}$ inch, valve travel 3 inches, steam lap (h. e. and c. e.) $\frac{7}{8}$ inch, exhaust lap (h. e. and c. e.) $\frac{3}{16}$ inch. Let $\frac{R}{L}$, that is, the ratio of the length of the crank to the connecting rod, equal $\frac{1}{5}$. Construct the Zeuner diagram and find all the events for both the head and crank ends in per cents.

Solution. Construct the valve travel circle $A\ C\ D\ F$, Fig. 26, with a radius of $1\frac{1}{2}$ inches; the steam-lap circle with a radius $O\ H$ of $\frac{7}{8}$ inch; and the exhaust lap circle with a radius $O\ R$ of $\frac{3}{16}$ inch. The steam-lap circle cuts the crank position for h. e. dead center at the point K. From K lay off the distance $J\ K$ to represent the lead of $\frac{1}{16}$ inch. At A, construct the lead circle with a radius of $\frac{1}{16}$ inch. From the properties of the Zeuner, we know that where a perpendicular erected at the lead point J cuts the valve travel circle as at B, the line $B\ O$ is the diameter of the valve circle and the angle $C\ O\ B$ is the required angle of advance. We also know that a line drawn perpendicular to $B\ O$ and tangent to the steam-lap circle cuts the valve travel circle at the points of admission and cut-off, respectively. Therefore, draw $S\ T_2$ so it will be tangent to the steam-lap circle and perpendicular to $B\ O$ at H. The points S and T_2 are the points of head-end admission and cut-off, respectively. It is to be noted, also, that this line $S\ T_2$ is tangent to the lead circle, which fulfills another condition of the property of the Zeuner.

To locate the other events for the head and crank ends, draw lines perpendicular to $B\ O\ E$ and tangent to the steam- and exhaust-lap circles, and the points where these lines cut the valve travel circles will be the required

points. In the same manner, the several other points in the figure have been located.

To find the per cent of stroke at which the several events occur, take a radius proportionately equal to the length of the connecting rod and describe the arcs shown. As $\frac{R}{L}$ is $\frac{1}{5}$, L equals 5 R. But R is one-half the valve travel, *i.e.*, 1.5 inches.

$$\therefore \quad L = 5 \times 1.5$$
$$= 7.5 \text{ inches}$$

Now, with a radius of 7.5 inches and with a center on the horizontal line through the center of the valve travel circle produced to the left of the

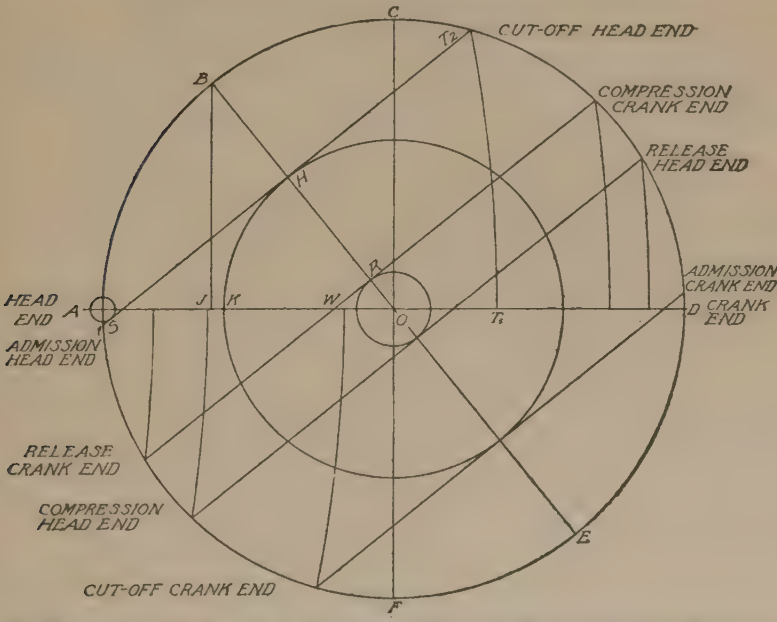

Fig. 26. Diagram for Finding Events for Head and Crank Ends; Lead, Valve Travel, and Laps being Given

vertical line $C\,F$, sweep the arcs shown from the points of admission, cut-off, etc., on the head and crank ends. Remembering that the head-end events are measured from the head-end dead center and the crank-end events from the crank-end dead center, measure the distance $A\,T_1$. This distance, 2.03 inches, divided by the valve travel 3 inches, and multiplied by 100, gives the per cent cut-off on the head end, that is,

$$\frac{2.03}{3} \times 100 = 67\% \text{ cut-off}_{\text{h.e.}}$$

In like manner, measure the distance $D\ W$ for the crank-end cut-off, which we find is 1.8 inches. Then

$$\frac{1.8}{3}\times 100 = 60\% \text{ cut-off}_{\text{h.e.}}$$

Continuing this procedure for the other events, the final results obtained from the diagram will be

Event	Head End	Crank End
Admission	98 per cent	98 per cent
Cut-off	67 per cent	60 per cent
Release	93 per cent	91 per cent
Compression	16 per cent	$12\frac{1}{2}$ per cent

Angle of advance $\theta = 40$ degrees

Example 3. Given an engine having 30 per cent cut-off on the head end; maximum port opening of $\frac{3}{8}$ inch; and lead on the head end $\frac{1}{16}$ inch. The laps are to be equal; compression on the head end is 25 per cent; and $\frac{R}{L}$ equals $\frac{1}{4}$. Construct the Zeuner diagram.

Solution. In Fig. 27, lay off $E\ F$ to represent the maximum port opening $\frac{3}{8}$ inch; $F\ G$ the lead $\frac{1}{16}$ inch; and erect perpendiculars $E\ J$, $G\ H$, $F\ I$. On any point as O on the line GH, draw a trial circle such as $A\ B\ C\ D$, which in this case was assumed to be 1 inch in diameter. Since cut-off on the head end occurs at 30 per cent of the stroke, locate the direction of the crank position $O\ P$ for this position. This direction will hold for any valve travel. Draw $O\ M$ perpendicular to $O\ P$, cutting $F\ I$ at K. Bisect the angle $F\ K\ M$ by $K\ N$. On $K\ N$ as a center line, find by trial a radius and center, such that a circle when described will pass through O and be tangent to $E\ J$. The center is found to be at L and the distance $O\ L$ is the radius of the required valve circle. With L as a center, draw a circle tangent to $F\ I$ and $K\ M$. Such a circle will be the required steam-lap circle. To demonstrate why this construction is correct, it is only necessary to refer to the properties of the Zeuner diagram as given in connection with Fig. 24. Here it is shown that a line drawn perpendicular to the crank position for the point of cut-off and tangent to the steam-lap circle cuts the valve travel circle at the extremity of the valve circle, as at B. Hence, $O\ M$ fulfills this condition, which gives the extremity of the required valve travel circle at O. In Fig. 24, it is also evident that the steam-lap circle is tangent to the perpendicular to the crank position for the given cut-off and is also tangent to a perpendicular to the horizontal center line drawn at the extremity of the maximum port opening. Therefore, this condition was fulfilled in establishing the required lap in Fig. 27. Having obtained the valve travel and lap, it remains to complete the diagram in order to determine the other conditions. In Fig. 28, the circles $A\ B\ C\ D$ and $a\ b\ c\ d$ are constructed on a diameter of $4\frac{13}{16}$ inches and $4\frac{1}{16}$ inches, respectively, the former being the value of the valve travel and the latter twice the steam lap, as found in Fig. 27. Locate the head-end cut-off at 30 per cent and draw the lead circle with a radius of $\frac{1}{16}$ inch. Locate the head-end compression of 25 per cent at I. Draw $G\ H$ tangent to the

steam-lap and lead circles cutting the valve travel circle at G, thus establishing the head-end admission. From the properties of the Zeuner diagram as discussed on pages 21 and 22, we know the diameter of the valve circles will be on a line bisecting the angle $G O H$. Draw the line $F O E$ bisecting this angle; this line will be perpendicular to $G H$. Having established $F O E$ and bear-

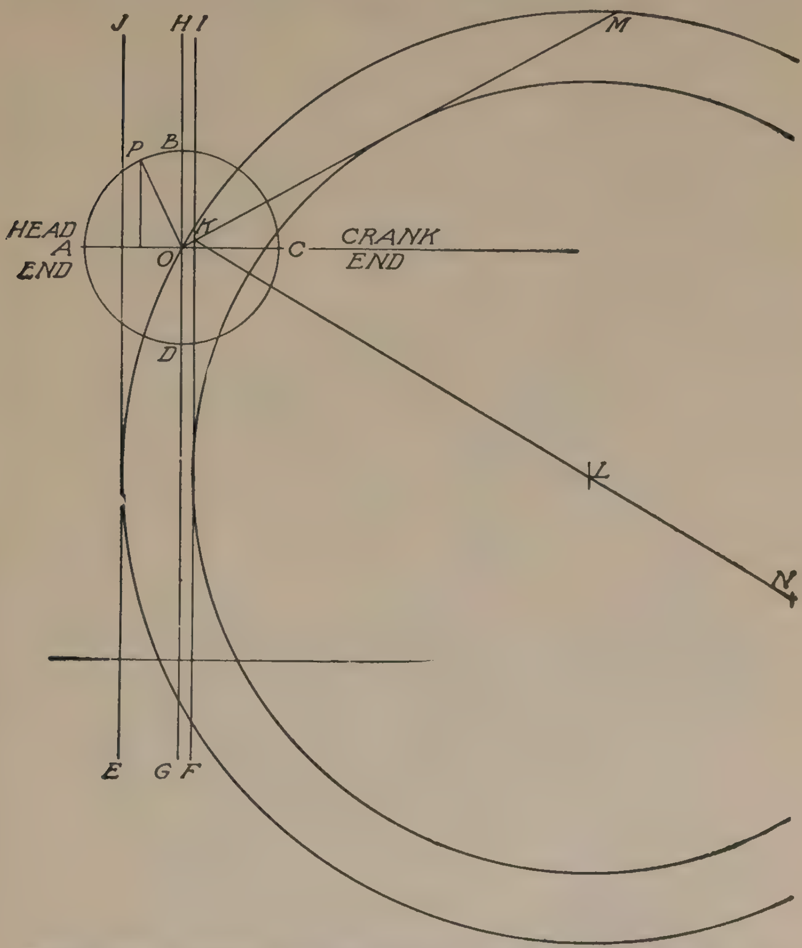

Fig. 27. Diagram for Engine with Thirty Per Cent Cut-Off, Laps Equal, Compression Twenty-Five Per Cent

ing in mind the demonstrations previously given, it is evident that a line drawn from the point of compression on the head end I perpendicular to $F O E$ will cut the valve travel circle at J, the point of head-end release. It is to be noted, however, that the line joining the points of release and compression on the head end lies on the same side of the center O as does the line joining the points

of admission and cut-off for the head end. This relation being opposite to that found in Fig. 24 means that instead of having exhaust lap with this valve, there is inside clearance equal to $O\,N$. With O as a center and $O\,N$ as a radius, describe the clearance circle and complete the Zeuner by drawing the parallel lines $K\,L$ and $P\,Q$, thus locating the remaining events of the stroke. In order to obtain the per cents of the events of the stroke, proceed as in Example 2.

The results are tabulated as follows:

Steam lap	$=2\frac{1}{32}$ inches
Inside clearance	$=\frac{5}{16}$ inch
Valve travel	$=4\frac{13}{16}$ inches
Angle of advance	=62 degrees
Admission on both h.e. and c.e.	=99 per cent (approx.)
Cut-off c. e.	=20 per cent
Compression c. e.	=18 per cent
Release c. e.	=60 per cent
Release h. e.	=72 per cent

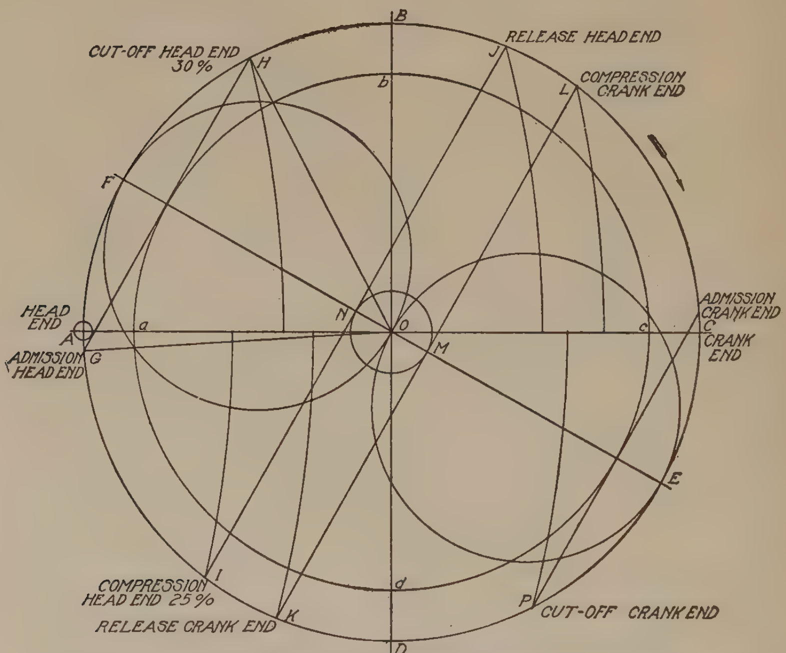

Fig. 28. Diagram for Example 3 to Determine Admission, Compression, and Release at Crank and Head Ends

The preceding problems involve nearly all of the properties of the Zeuner diagram and, if completely mastered by the student, should make the solution of other problems very much easier.

Effect of Changing Lap, Travel, or Angular Advance. We are now in a position to consider more in detail the effect of changing

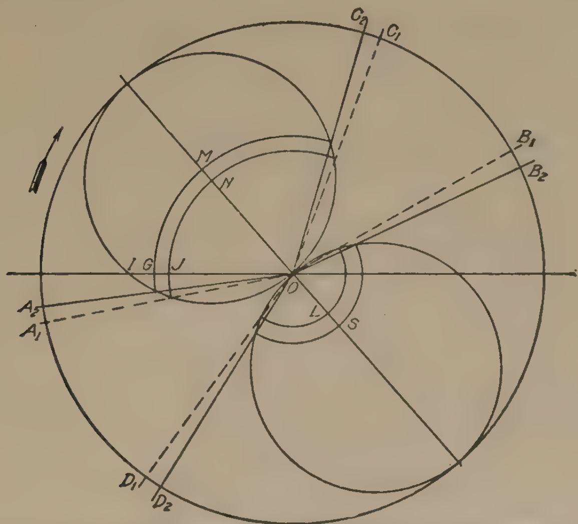

Fig. 29. Study of Effect of Changing Valve or its Setting

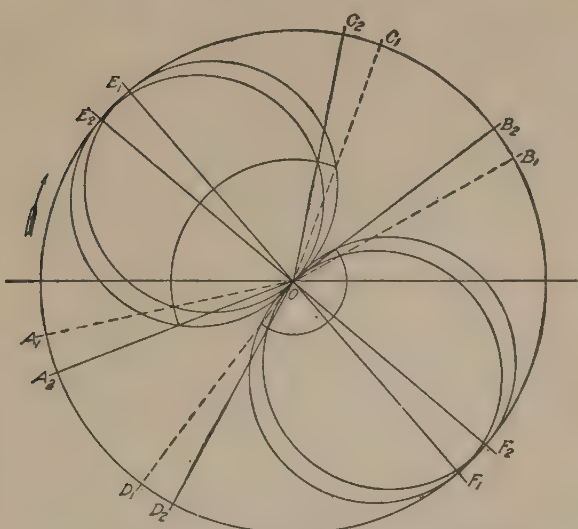

Fig. 30. Study of Effect of Changing Angle of Advance

in any way either the valve or the setting. Let us consider **Fig.** 29, which is in every way like Fig. 23 except that all unnecessary

TABLE I

Effect of Changing Lap, Travel, and Angular Advance

Event	Increasing Outside Lap	Increasing Inside Lap	Increasing Travel	Increasing Angular Advance
Admission	Is later Ceases sooner	Not changed	Begins earlier Continues longer	Begins earlier Same period
Expansion	Is earlier Continues longer	Beginning unchanged Continues longer	Begins later Ceases sooner	Begins earlier Same period
Exhaust	Unchanged	Occurs later Ceases sooner	Begins earlier Ceases later	Begins earlier Same period
Compression	Begins at same point Continues longer	Begins sooner Continues longer	Begins later Ceases sooner	Begins earlier Same period

letters and lines are omitted to avoid confusion. If the outside lap, or steam lap, is *increased* an amount equal to $N M$, the admission will take place later, viz, at crank position $O A_2$; the lead will be reduced to $I G$ and cut-off will take place earlier, viz, at $O C_2$.

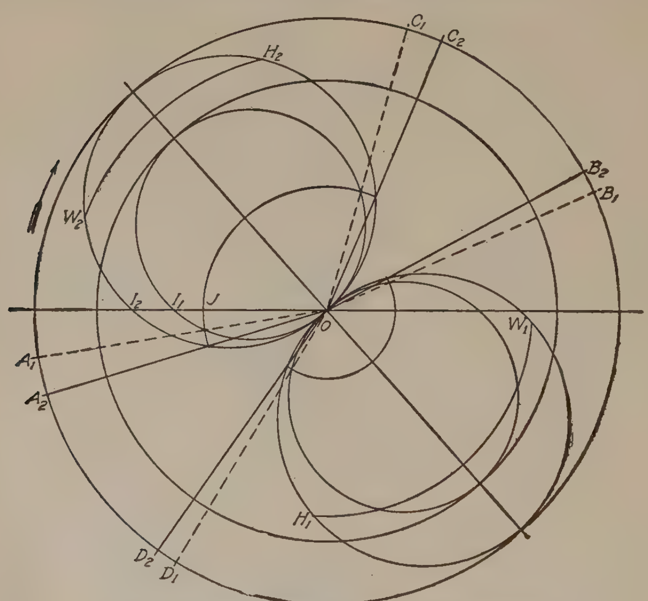

Fig. 31. Study of Effect of Changing Eccentricity

If the outside, or steam lap, is *reduced* a like amount, the contrary effects will be observed. If the inside lap, or exhaust lap, is increased an amount equal to $L S$, the release will take place later at the crank position $O B_2$, and compression will take place earlier at $O D_2$. The

contrary effect will be observed by decreasing the inside lap, or exhaust lap.

If the angular advance is increased, all the events will occur earlier, as is evident from Fig. 30. The crank revolves in the direction indicated by the arrow and $O\,A_2$ (new position of admission) is *ahead* of $O\,A_1$ the old position.

If the eccentricity is increased, Fig. 31, the valve travel will increase and admission will take place earlier at $O\,A_2$; the lead will be increased an amount equal to I_1I_2, and cut-off will take place later at $O\,C_2$. Release will be earlier at OB_2 and compression will be later at $O\,D_2$. The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W_2 and H_2, and the admission port will be open wide while the crank is moving from $O\,W_2$ to $O\,H_2$. Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W_1 and H_1. The steam port is then wide open to exhaust while the crank is moving from W_1 to H_1. From the above, it will be seen that the periods are all changed by changing the travel, thus admission and exhaust begin sooner and last longer, and expansion and compression begin later and cease sooner.

For convenience, these results are collected in Table I, which shows the effect of changing the laps, travel, and angular advance.

There are, of course, all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the problems solved above.

DESIGN OF SLIDE VALVE

In designing a slide valve, some of the variables are assumed and the others are found by means of the diagrams presented above. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Port. *Steam Supply Pipe.* It is generally conceded by authorities that the pipes supplying steam to steam engines should be of such dimensions that the mean velocity of steam in them would not exceed 6,000 feet per minute. If the velocity of steam exceeds 6,000 feet per minute, there will be a very appreciable

loss of pressure, which is objectionable. In computing the size of a steam supply pipe for an engine, the assumption is made that the cylinder is filled at each stroke. The volume of steam passing through the steam pipe must equal the total volume of steam used by the cylinder.

Let d equal diameter of steam pipe in inches; D equal diameter of cylinder in inches; L equal length of stroke in feet; and N equal revolutions per minute (r. p. m.).

The area of the steam pipe in square feet would be $\frac{\pi d^2}{4\times 144}$ and that of the cylinder would be $\frac{\pi D^2}{4\times 144}$. The total volume of steam flowing through the pipe per minute would be $\frac{\pi d^2}{4\times 144}\times 6000$. Disregarding the volume of the piston rod, the total volume of steam used by the cylinder in one minute would be $\frac{\pi D^2}{4\times 144}\times 2LN$.

Since the volume of steam flowing through the pipe per minute must equal that used by the cylinder in the same time, we can equate the two expressions; that is,

$$\frac{\pi d^2}{4\times 144}\times 6000 = \frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2 = \frac{D^2LN}{3000}$$

$$d = \frac{D\sqrt{LN}}{54.772}$$

Exhaust Pipe. For exhaust pipes, the mean velocity of steam is taken as 4,000 feet per minute. Therefore

$$\frac{\pi d^2}{4\times 144}\times 4000 = \frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2 = \frac{D^2LN}{2000}$$

$$d = \frac{D\sqrt{LN}}{44.721}$$

Example. Suppose an engine is 10 inches × 18 inches, and makes 180 revolutions per minute. Determine the diameters of the steam and exhaust pipes.

Solution. Substituting in the equation

$$d = \frac{D\sqrt{LN}}{54.772}$$

gives for the diameter of the steam supply pipe

$$d = \frac{10\sqrt{1.5 \times 180}}{54.772}$$
$$= \frac{164.3}{54.772}$$
$$= 3 \text{ inches}$$

The required diameter of exhaust pipe would be

$$d = \frac{D\sqrt{LN}}{44.721}$$
$$= \frac{10\sqrt{1.5 \times 180}}{44.721}$$
$$= \frac{164.3}{44.721}$$
$$= 3.67 \text{ inches}$$

A 4-inch pipe would probably be used.

In practice different builders use different formulas, but all are derived from the fundamental assumptions made above, with certain constants added for different types of engines. The size of both steam and exhaust pipes required for engines of the same class is not affected in any marked degree by different types of valve gears.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of six-tenths to nine-tenths of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about eight-tenths the diameter of the cylinder. Then in the 10-inch $\times$ 18-inch engine, the steam ports would be $.8 \times 10$, or 8 inches long. If the area for admitting steam is 7.0686 square inches (corresponding to a pipe 3 inches in diameter) and the length of port is 8 inches, the width will be $\frac{7.0686}{8}$, or .8836 inch—about $\frac{7}{8}$ inch.

The width of port opening would be about $.9 \times .8836$, or .79524 inch—about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port, as shown in Fig. 7, reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large, for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that the outside edges of the valve can not uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should not be less than the thickness of the cylinder wall in order to make a good casting.

Point of Cut-Off. In the study of Steam Engine Indicators, it was shown that if the point of cut-off is too early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the point of cut-off is controlled either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression, which at a very early cut-off may be excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut off earlier than one-half or two-thirds stroke. If an earlier cut-off is desired, large outside laps are necessary.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine and type of valve gear. An engine having high compression that compresses the steam nearly to boiler pressure will give good results with little or no lead. If the ports are small and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

ILLUSTRATIVE PROBLEM

EXAMPLE. Design and lay out the valve and valve seat for an engine of cylinder diameter 10 inches, stroke 18 inches, revolutions 180 per minute, lead angle 3 degrees, cut-off equal at both ends and taking place at 75 per cent of stroke, maximum port opening .9 area of steam pipe, compression 15 per cent of the stroke at both ends, and length of connecting rod 3 feet.

SOLUTION. The piston displacement, or cylinder volume, will be $\frac{3.1416\times10^2}{4}\times18=1413.7$ cubic inches, or .818 cubic feet.

If the engine makes 180 revolutions, neglecting the volume of the piston rod, it will use $2\times180\times.818=294.48$ cubic feet of steam per minute. Steam pipe area $=\frac{294.48}{6000}=.0491$ square feet, or 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made eight-tenths the diameter of cylinder, the width will be $\frac{7.07}{8}=.88$ inch, or about $\frac{7}{8}$ inch. The width of maximum port opening will be $.9\times.88=.792$ inch, or nearly $\frac{13}{16}$ inch.

Zeuner Diagram. It will be necessary to draw a separate valve circle for each end of the cylinder. First, consider the head end. The valve travel not being known, we shall lay off $X\ Y$ on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 32. Lay off the lead angle $X\ O\ A_1=$ 3 degrees. Lay off $X\ C_2=.75$ of the assumed valve travel $4\frac{1}{2}$ inches. Draw the arc C_1C_2, as previously explained, and draw $O\ C_1$ which will be the crank position at the point of cut-off. The radius of the arc C_1C_2 will be equal to four times the radius of the eccentric circle, or 12 inches, because the connecting rod is four times the length of the crank. Let the line $O\ E_1$ bisect the angle $A_1O\ C_1$, and on $O\ E_1$ draw the valve circle. $O\ V_1$ $(=O\ K_1)$ is then the outside lap, with these assumed conditions. Drawing the lap circle, the maximum port opening E_1N_1 is found to equal $1\frac{7}{16}$ inches, although $\frac{13}{16}$ is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity is found from the proportion

$$x:3::\tfrac{13}{16}:1\tfrac{7}{16}$$

$$x=1\tfrac{11}{16}\text{ inches}$$

Now draw a new eccentric circle with a radius of $1\frac{11}{16}$ inches and a new valve circle with a diameter $O\ E_2$ $1\frac{11}{16}$ inches. $O\ K_2$ is now the outside lap and

the maximum port opening is equal to $E_2 N_2$, which from actual measurement is found to be $\frac{13}{16}$ inch. The outside lap $O K_2$ ($=O V_2$) is $\frac{27}{32}$ inch and the lead $I J$ is $\frac{3}{32}$ inch.

Produce E_1O to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off $O H_2 = O C_2$ and, with a radius of 12 inches, draw the arc H_1H_2. Then, as already explained, $O H_1$

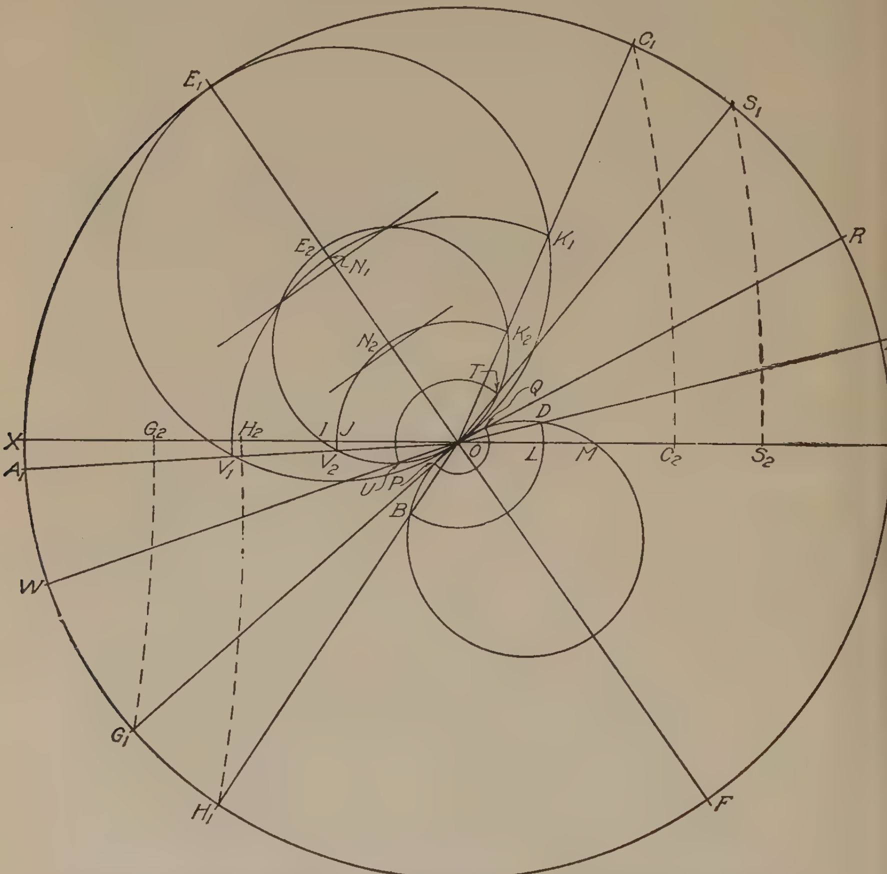

Fig. 32. Zeuner Diagram for Design of Valve and Valve Seat in Problem Page 35

will be the crank angle at cut-off on the return stroke. $O B$, the outside lap, will be $\frac{19}{32}$ inch. Draw the lap circle intersecting the valve circle at D. Then $O D A_2$ is the crank position at admission on the return stroke and $L M$, $\frac{3}{8}$ inch is the lead on the crank end of the cylinder. The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

Now lay off $X G_2$ equal to fifteen-hundredths of $X Y$ and find the crank position $O G_1$. This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to $O P$. Draw the lap circle $P Q$, which allows us to draw through Q the crank line $O R$, which is the release on the forward stroke.

Lay off $Y S_2$ $(= X G_2)$ equal to fifteen-hundredths of $X Y$, and construct the crank line $O S_1$, which is the crank position at the crank-end compression. $O S_1$ intersects the valve circle at T, giving $O T$, $\frac{7}{16}$ inch, as the inside lap on the crank end. Draw this lap circle, which will intersect the valve circle at U. This enables us to draw $O U W$, the crank position at release, on the return stroke.

Layout of Valve. From the data determined by means of these diagrams, the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

DATA	HEAD END	CRANK END
Cut-off (per cent of stroke)	75 per cent	75 per cent
Outside lap	$\frac{27}{32}$ inch	$\frac{19}{32}$ inch
Inside lap	$\frac{7}{32}$ inch	$\frac{7}{16}$ inch
Lead	$\frac{3}{32}$ inch	$\frac{3}{8}$ inch
Port opening	$\frac{13}{16}$ inch	$1\frac{1}{16}$ inches
Width of port	$\frac{7}{8}$ inch	$\frac{7}{8}$ inch

Fig. 33 shows this valve in section. Let us begin at the end having the largest inside lap or, in this case, at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide and the crank-end outside lap $\frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap $\frac{7}{16}$ inch. When the valve moves to the left, the point E_2 will travel $1\frac{11}{16}$ inches—a distance equal to the eccentricity—and in this position of extreme displacement, the exhaust port E_1F must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore, we lay off $E_1 F$ equal to $1\frac{11}{16}$ inches + $\frac{7}{8}$ inch = $2\frac{9}{16}$ inches. The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end outside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

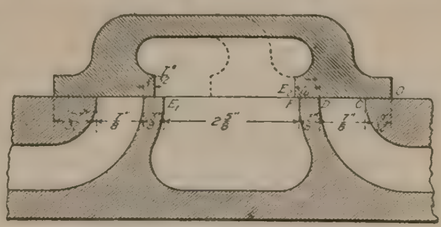

Fig 33. Section of Valve Designed from Diagram Fig. 32

Reversing Simple Engine. In the operation of a simple engine having a plain slide valve or a piston valve, it sometimes becomes necessary to reverse the direction of rotation of the engine shaft. Remembering the principles presented in the foregoing study of the Zeuner diagram, this is not a difficult task.

It is proposed to here show *first,* how an engine may be reversed with a direct valve, engine running over; *second,* with a direct valve, engine running under; *third,* with an indirect valve, engine running over; and *fourth,* with an indirect valve, engine running under.

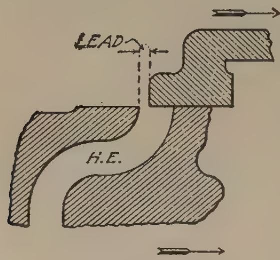

Fig. 34. Section Showing Lead of Valve, Engine Running Over

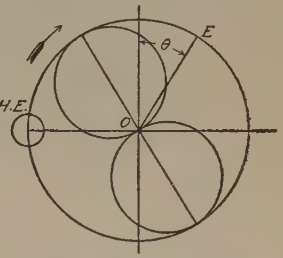

Fig. 35. Diagram for Direct Valve, Engine Running Over

Definitions. Before explaining the operation for obtaining the above, it is well to have an understanding of the meaning of the terms "direct" and "indirect" as applied to a valve, and of "running over" and "under" as applied to an engine.

A valve is said to be a *direct,* or *outside admission, valve,* when at the beginning of the stroke the valve and the piston are moving in the same direction, as indicated by the arrows in Fig. 6. It is

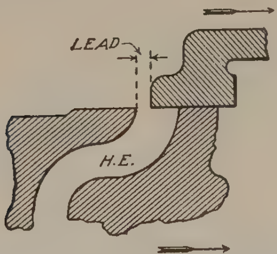

Fig. 36. Section Showing Lead for Direct Valve, Engine Running Under

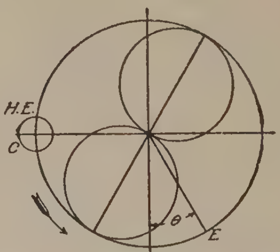

Fig. 37. Diagram for Direct Valve, Engine Running Under

also to be noted that steam is being admitted to the cylinder by the outer edge of the valve, which is the reason for calling it an outside admission valve.

If, in Fig. 6, the valve should be moving in the opposite direction from that shown and steam should be entering the cylinder by the inner edge of the valve, the valve would then be said to be an *indirect,* or *inside admission, valve.*

Most plain slide valves are of the outside admission type, while most piston valves are of the inside admission type.

An engine is said to be *running over*, if, when the piston is moving from the head end toward the crank end, the moving parts, such as connecting rod, crank, etc., are above the center line, as shown in Fig. 17. The engine is said to be *running under* when the above mentioned parts are below the center line when the piston is moving from the head end toward the crank end.

Direct Valve, Engine Running Over. In Fig. 34, let the valve have lead equal to that shown. Since this is a direct valve, engine running over, the valve will be to the right of its mid-position and moving to the right, hence the eccentric will be $(90+\theta)$ degrees ahead of the crank. If the engine is on the head-end dead center, the eccentric would be at E, that is, $(90+\theta)$ degrees ahead of the crank. The right and left valve circles will be located in the second and fourth quadrants, respectively, as shown in Fig. 35.

Direct Valve, Engine Running Under. With a direct valve, engine having lead and running under, as illustrated in Figs. 36 and 37, the valve will be in the same relative position as in the former case, when the crank is on the head-end dead center. In this position the valve must be to the right of its mid-position and moving towards the right, hence the eccentric must be, as shown at E, Fig. 37, an angular distance of $(90+\theta)$ degrees ahead of the crank.

The right and left valve circles will be located in the first and third quadrants, respectively, as shown in Fig. 37.

It is to be noted on comparing the position of the eccentric in Figs. 35 and 37 that both of the eccentric positions make an angle equal to the angle of advance with the vertical. Therefore, to reverse a direct valve, engine running over, turn the eccentric around the shaft, in the direction in which the engine is running, by an angle of $(180-2\theta)$ degrees, or turn the eccentric ahead of the crank, in the direction in which the engine is to run, an angle of $(90+\theta)$ degrees.

Indirect Valve, Engine Running Over. An indirect valve engine running over is illustrated in Figs. 38 and 39. Remembering that the valve must be moving to the left as the piston moves from the head end toward the crank end, and that the valve must be displaced by an amount equal to the lap plus the lead to the left of its mid-position, the eccentric must be below the horizontal and behind

the crank an angular distance of $(90-\theta)$ degrees. Hence, it is located at E, Fig. 39. The right and left valve circles will be located in the fourth and second quadrants, respectively

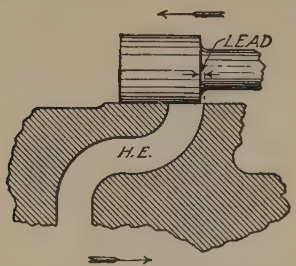

Fig. 38. Section of Indirect Valve, Engine Running Over

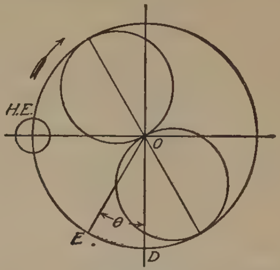

Fig. 39. Diagram of Indirect Valve, Engine Running Over

Indirect Valve, Engine Running Under. To locate the eccentric for an indirect valve engine having lead and running under (see Figs. 40 and 41), proceed as before. The eccentric will be found at E, Fig. 41, and the right and left valve circles will be located in the first and third quadrants, respectively.

An examination of Figs. 38 to 41 will disclose the fact that to reverse an engine using an indirect valve, it is only necessary to turn the eccentric through an angle of $(180-2\theta)$ degrees in the direction in which the engine shaft is turning or, in other words, the procedure is the same as for a direct valve.

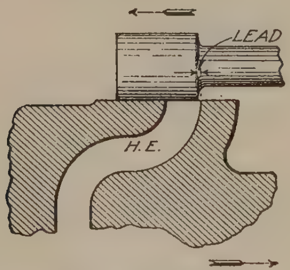

Fig. 40. Section of Indirect Valve, Engine Running Under

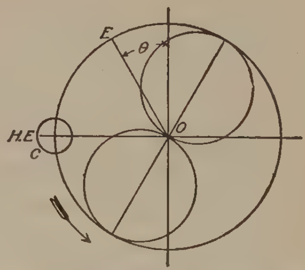

Fig. 41. Diagram for Indirect Valve, Engine Running Under

Comparisons and Comments. A comparison of Figs. 34 and 35 with 38 and 39 will indicate the relative positions of the eccentric for an engine running over with a direct valve and for one running over with an indirect valve. It is evident that in the first case, the eccentric precedes the crank by an angle of $(90+\theta)$ degrees, whereas

in the second, the eccentric follows the crank by an angle of $(90-\theta)$ degrees. This same condition is true for two engines running under, one using a direct valve and the other an indirect.

As an aid in locating the valve travel circles after the eccentric position has been determined, remember that the quadrant separated by a vertical line through the center, from the quadrant containing the eccentric position, is the quadrant in which the right valve travel circle is to be located.

All of the study on the Zeuner valve diagram thus far has to do with an engine running over having a direct valve. After the location of the eccentric position has been determined for the above various conditions, the construction of the Zeuner diagram should be a simple matter.

The principles underlying the location of the eccentric for an engine running over or under and having a direct or indirect valve should be borne in mind when setting valves.

VALVE SETTING

Possible Adjustments. The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken, the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps can not be altered without considerable labor, and the throw or eccentricity of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put Engine on Center. It is usual to put the engine on center before setting the valve. First, put the engine in a position where the piston has nearly completed the outward stroke and make a mark M_1, Fig. 42, on the guide opposite the corner of the crosshead at some convenient place. Also make a mark P with a center punch on the frame of the engine near the crank disk. With

this mark P as a center, describe an arc C on the wheel rim with a tram.*

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead and make another

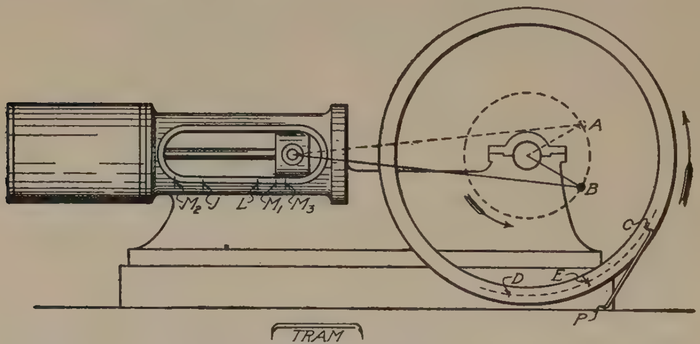

Fig. 42. Sketch of Engine, Showing Method of Putting Engine on Center

mark D on the wheel with the tram, keeping the same center. With the center of the pulley, or crank disk, as a center, describe an arc CD on the rim, which intersects the two ares drawn with the tram. Bisect the arc CD and turn the engine until the new point is distant from the point P an amount equal to the length of the tram, in which position the engine will be on center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin

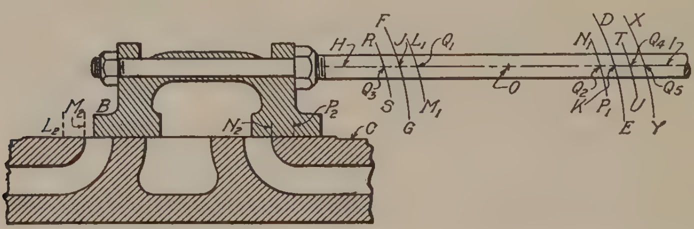

Fig. 43. Diagram Showing How Valve is Set for Equal Lead

will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point while the engine is moving in the proper direction. In this manner, the dead center can be located for both the head and crank ends.

*A tram is a steel rod with its ends bent at right angles and sharpened.

To Set Valve for Equal Lead. After locating the dead center points as described above the next step is to locate what are known as the port marks. In Fig. 43 move the valve to the left until cut-off occurs on the head end or until the edge of the valve at B is at M_2. Then, with a center C on some fixed point on the cylinder or engine frame, describe with a tram the arc $F\ G$ on the valve rod. Continue the rotation of the engine in the same direction until cut-off takes place at the crank end. Then with the same tram and center C, sweep the arc $D\ E$ on the valve rod. Draw the center line $H\ I$ and where this center line cuts the arcs $F\ G$ and $D\ E$, mark the points J and K, respectively, which points are known as the port marks. Bisect the distance between J and K, thus establishing the point O. When one tram point is in C and the other just enters the point J, the valve is just cutting off on the head end; and when the tram point coincides with C and K, it is an indication that cut-off is occurring on the crank end, hence a basis of comparison has been established for the two ends. Place the engine on the forward dead center and sweep the arc L_1M_1. The distance between the arcs L_1M_1 and $F\ G$, which is equal to JQ_1, represents the amount the valve extends over the port when the engine is on the head-end dead center. In a like manner, establish the arc N_1P_1 when the engine is on the crank-end dead center, in which position the valve overlaps the steam port the distance KQ_2. In order to have equal travel of the valve on either side of its mid-position, the distance $J\,Q_1$ should equal $K\,Q_2$. If necessary to equalize these distances, lengthen or shorten the valve stem as required. Having secured an equal valve travel, place the engine on the forward dead center. Since the engine is running under (see Fig. 37), the eccentric will be found $(90 + \theta)$ degrees ahead of the crank in the direction the engine is to run. Lay off on the valve stem the distances $J\,Q_3$ and $K\,Q_4$ equal to the required lead. With the tram point in C and the engine placed on the head-end dead center, turn the eccentric in the direction in which the engine is to run—which is *under* in this case—until the arc $R\ S$ passes through the point Q_3. Fasten the eccentric and turn the engine around until it is on the crank-end dead center. Sweep another arc as $T\ U$ with the tram. If this arc passes through the point Q_4, then the valve is correctly set for equal lead, that is, JQ_3 is equal to KQ_4. If, however, the arc $T\ U$

does not pass through the required point Q_4, but falls beyond, it is an indication of unequal lead, so a correction must be made. Suppose, for instance, that when the crank was placed on the crank-end dead center, the arc described from C fell at XY instead of TU, then it is obvious that the crank end has more lead than the head end. To make a correction for this inequality, find the difference between the lead on the head and crank ends—which in this case is equal to the distance Q_4Q_5—and correct half of the difference on the valve stem and the other half by altering the angle of advance. In this case, the valve stem should be lengthened by the amount $\frac{Q_4Q_5}{2}$, which would increase the lead on the head end by that amount and decrease it by the same amount on the crank end. After establishing an equal travel of the valve by adjusting the length of the valve stem, thus giving an equal amount of lead at each end, the desired amount of lead may be obtained by changing the angle of advance. To obtain the required lead in this case, it would be necessary to reduce the angle of advance. It may be necessary to make several trials before the desired results are obtained, this being particularly true if working on an engine having lost motion in the various parts. In order to eliminate the effect of lost motion in so far as possible, the engine should always be turned in the direction which it is to run.

In case it is difficult to turn an engine, the following method may be used. First, loosen the eccentric on the shaft and turn it around until it gives a maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set Valve for Equal Cut-Off. To set the valve for equal cut-off, observe the preliminary steps of locating on the valve stem the dead-center points, port marks, and equal travel of the valve to either side of its mid-position, as described in connection with setting the valves for equal lead.

Assume that it is desired to set the valves for an equal cut-off of 75 per cent. On the guides of the engine illustrated in Fig. 42,

locate the points M_2 and M_3, corresponding to the extreme positions of the edge of the crosshead, or a given point on the crosshead. The distance M_2M_3 represents the stroke of the piston, so when 75 per cent cut-off occurs, the reference point on the crosshead should be at a point J, which is 75 per cent of the stroke M_2M_3 for the crank end and at the point L for 75 per cent cut-off on the head end. Remembering that the points J and K on the valve stem in Fig. 43 represent points of cut-off, all required reference points needed are known. Turn the engine over in the direction indicated in Fig. 42 until the reference point on the crosshead corresponds to the reference point on the guide, as L, for the head-end cut-off. Then with the tram in the center C, Fig. 43, describe an arc, say, L_1M_1. Continue the rotation of the engine in the same direction until the piston has completed the forward stroke and has returned to the point where the reference lines on the crosshead and the guide J coincide. Tram the valve stem as before, locating the arc, say, N_1P_1. Since the tram should coincide with the arcs $F\,G$ and $D\,E$ for the head-end and crank-end cut-off, respectively, it is therefore evident that with the tram coinciding with L_1M_1 and N_1P_1 that the required cut-off is not obtained but occurs too early. Since the distances Q_1J and Q_2K are equal, the length of the valve stem does not need to be disturbed. To make cut-off occur later, decrease the angle of advance by moving the eccentric opposite to the direction in which the engine is to run. For instance, with the engine standing so that the point L, Fig. 42, and the end of the crosshead are coincident, move the eccentric until the tram points coincide with C and J, Fig. 43. Try the points for the cut-off on the crank end, and if the tram fits easily into C and K, then the valve is set correctly. If, however, the tram points do not fit into the points C and K, continue the operation until the desired results are obtained.

From the above discussion, two points have been established:

(1) Moving the valve on the valve rod changes the corresponding events the same on both ends, one being made earlier and the other later. That is, if the cut-off is made earlier on the head end, it will be later on the crank end, and so on for the other events.

(2) Moving the eccentric on the shaft or changing the angle of advance changes the corresponding events the same for both ends, both being made earlier or both later.

MODIFICATIONS OF THE SLIDE VALVE

Balancing Steam Pressure. The ordinary slide valve is most suitable for small engines. For engines of large size, some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of

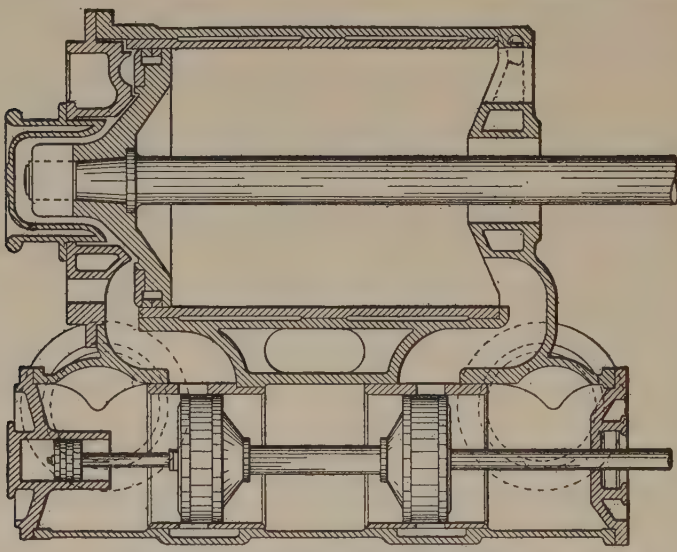

Fig. 44. Section of Piston Valve and High-Pressure Cylinder of U. S. S. "Massachusetts" Showing Method of Balancing

power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or *vice versa*. With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement, the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action

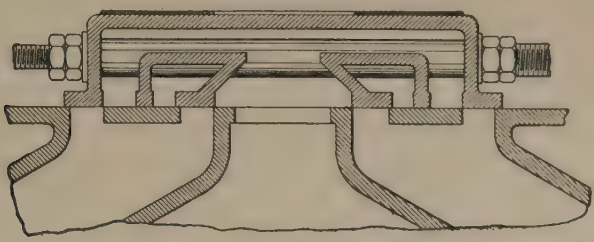

Fig. 45. Section of Double-Ported Slide Valve

is the same except that it is balanced so far as the steam pressure is concerned, the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 44 shows a section of the piston valve ana the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke,

especially those having a plain slide valve with short travel. This difficulty may be overcome by means of the double-ported valve shown in Fig. 45. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except there is communication between its exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

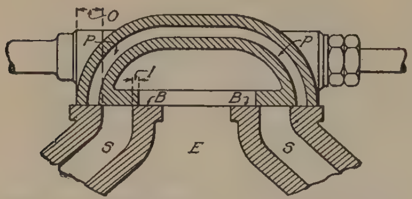

Fig. 46. Trick Valve Shown in Mid-Position

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true and fit so that steam is prevented from acting on the back of the valve.

Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve,

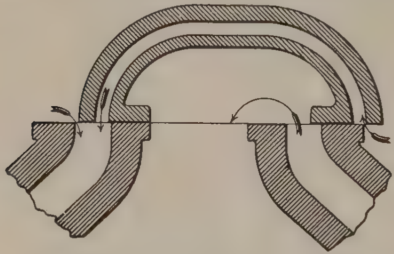

Fig. 47. Trick Valve Showing Admission of Steam Just Beginning

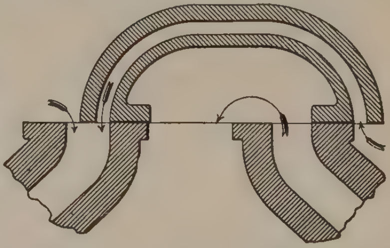

Fig. 48. Trick Valve at Extreme Right Position with Steam Port Open Wide

which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

In Fig. 46 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage *P P* through it. It has an outside lap *O* and an inside lap *I*. The seat is raised and has steam ports *S S*, bridges *B B*, and exhaust port *E*. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 47. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve, it will be admitted through the passage as shown in Fig. 47. If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is, $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

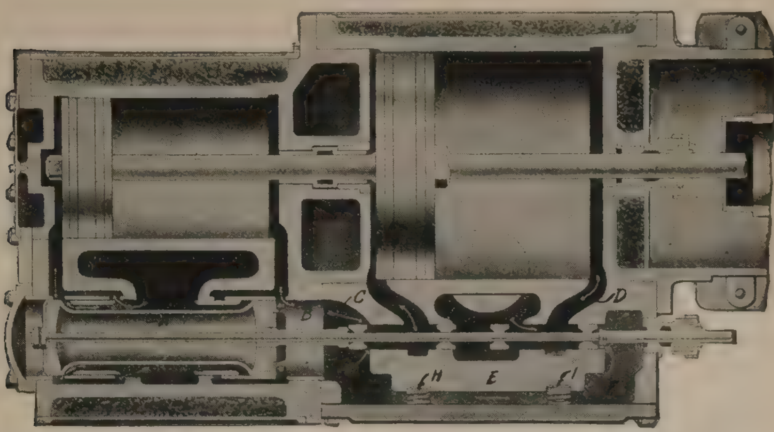

Fig. 49. Obtaining Perfect Balance by Use of Double-Ported Piston Valve and Double-Ported Slide Valve in Compound Engine

Fig. 48 shows the valve when it is in extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission, that used with the Armington and Sims engine being, perhaps, the best example.

Application of Various Types. Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

The double-ported valve, the trick valve, and others, often have some device for relieving the pressure, such as a bronze ring or cylinder fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

Fig. 49 well illustrates the application of a double-ported piston valve and a double-ported slide valve to a compound engine. It also shows a method used for obtaining a perfect balance. The piston valve on this engine is a hollow inside admission valve. The steam passes from the cavity *A* through the double ports in the piston valve, forcing the high pressure piston to the right, which action causes the exhaust steam to pass out of the high pressure cylinder through the passage *B* into the steam chest of the low pressure cylinder. The steam passes around the flat valve at *C C* into the low pressure cylinder. The steam back of the low pressure piston passes through the port *D* into the exhaust cavity. The pressure plate *E* is held against the flat slide valve by the springs *H* and *I*, there being steam all around the pressure plate, as at *F* and *G*. The valve fits closely between the valve seat and pressure plate, but the pressure plate being supported at the sides eliminates the pressure between the valve and its seat. Both of these valves are said by the builders to be in perfect balance.

Reversing Mechanism. In the early development of valve gears, it became necessary to devise some means of reversing the engine, hence it is found that a great many of the most prominent gears, such as the Stephenson, Walschaert, Marshall, and many others of more or less merit, embody the reversing feature.

Reversing by Means of One Eccentric. At first, the reversing of an engine was accomplished by the use of one eccentric. there being two methods by which this was done.

(1) The device shown in Fig. 50 was used on some of the earliest locomotives and marine engines, and may now be found as the reversing medium for engines used on small launches. The eccentric *E* is loose on the shaft between a fixed collar *G* and a hand wheel *H*. A stud projecting from the eccentric passes through a curved

slot in the disk of the wheel and can be clamped by a hand nut F. When running forward with the crank at C, the center of the eccentric is at A and the nut is clamped at F. To reverse, steam is shut off and, when the engine stops, the nut F is loosened and then moved to B and clamped; or, after F is loosened, the wheel, shaft, crank, and propeller are turned over by hand until B strikes the stud at F, where it is clamped. The engine will then run astern.

(2) The eccentric was mounted on a sleeve, which could be moved longitudinally along the shaft of the engine by means of a lever. The sleeve had a spiral slot cut on the inside of it, which subtended an angle of $(180-2\theta)$ degrees. This slot fitted over a radial pin on the shaft, so when the sleeve was pushed in or out by the lever, both the sleeve and the eccentric were turned through $(180-2\theta)$ degrees, thus reversing the engine.

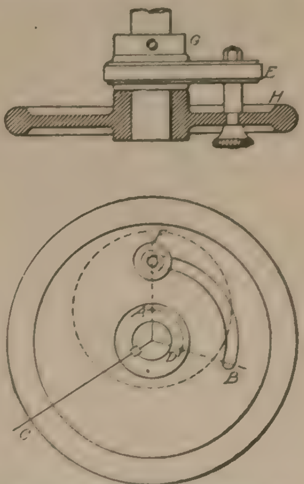

Fig. 50. Early Reversing Device by Means of One Eccentric

Reversing by Means of Two Eccentrics and Gab-Hooks. It is obvious from the foregoing that the method of reversing by shifting one eccentric is awkward and not well adapted to high speeds and large engines. It was a natural transition, therefore, from the one eccentric to the more convenient reversing gears having two eccentrics, one set $(90+\theta)$ degrees ahead of the crank for the forward motion and one $(90+\theta)$ degrees behind the crank for the backward motion. At first, this arrangement was rather crude and objectionable in some respects, as will be noted later. The essential feature to be borne in mind with reference to a two-eccentric gear is, that the object is to have only one eccentric at a time operating the valve. In the early development, this was accomplished by using gabs or gab-hooks, which could be brought in contact with the valve rod at the pleasure of the operator. For instance, if the engineer wished to go forward, he would lower the arm R, Fig. 51, thus bringing B_1 in contact with the valve rod at V. The valve would then be operated by the forward eccentric E_1 and the engine would run

in the direction indicated by the arrow at E_1. To reverse the engine, B_1 would be disengaged and B_2 placed in connection with V. The valve would then be operated by the eccentric E_2, and the engine would run in the direction indicated by the arrow at E_2, which is the reverse of that indicated at E_1. It is to be particularly noted that only one eccentric actuates the valve at one time. All reversing gears of the two-eccentric type carry out this principle to a greater or less extent.

It will be obvious that the gab-hooks are an improvement over the shifting eccentric, but even they have certain objectionable features, the three principal ones being (1) the engine must have a slow speed of rotation; (2) the engine must be of such construction that

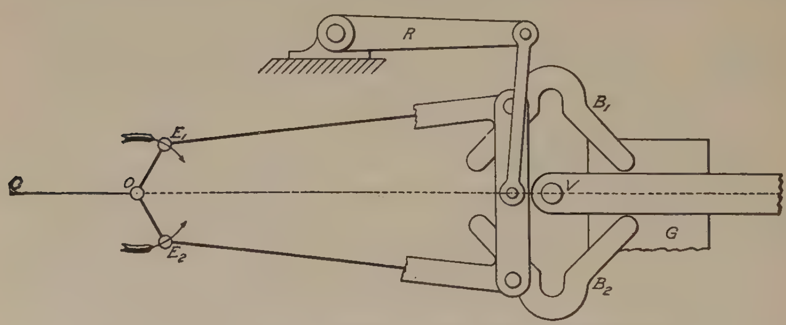

Fig. 51. Reversing Device Using Two Eccentrics and Gab-Hooks

the reversal can be accomplished in a leisurely manner—it is not convenient to reverse at a high speed with a gab-hook, but the engine must be turning slowly when the hook is dropped upon the pin; (3) the engine must be of such a type that it can be started by hand-working of the valves.

Reversing by Means of Two Eccentrics and Curved or Straight Links. To overcome these objectionable features, a step forward was taken when the gab-hooks were replaced by the curved or straight link, which is now used in connection with almost all reversing gears. This was a decided improvement as it not only accomplished the reversing of the engine but also made possible a variation in the adjustment of the valve mechanism, which permitted much more economical distribution of steam in the cylinder. There are two general classes of valve gears that use the curved link and its neces-

sary attachments, namely, the shifting link or the stationary link type, and the radial gear type.

The Stephenson gear is a worthy exponent of the shifting link type. The Walschaert, Joy, Marshall, and others are representatives of the radial gear type.

SHIFTING LINK TYPE OF VALVE GEAR

Stephenson Link Motion. As the Stephenson gear is one of the oldest reversing gears used and is perhaps the best known, a discussion of its principal features is presented first. This gear has been successfully used on stationary, traction, and marine engines, but its largest and, perhaps, most successful application has been on American locomotives. This gear is illustrated in Fig. 52. The two eccentrics E_1 and E_2, whose centers are at C_1 and C_2, respectively, are shown in their relative positions when the crank OA is at the crank-end dead center. The eccentric rods R_1 and R_2 are connected by forked ends to the link pins H and G. The link consists of two curved bars bolted together in such a manner that they may slide by the link block N. On the link are three sets of trunnions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links $F\,M$.

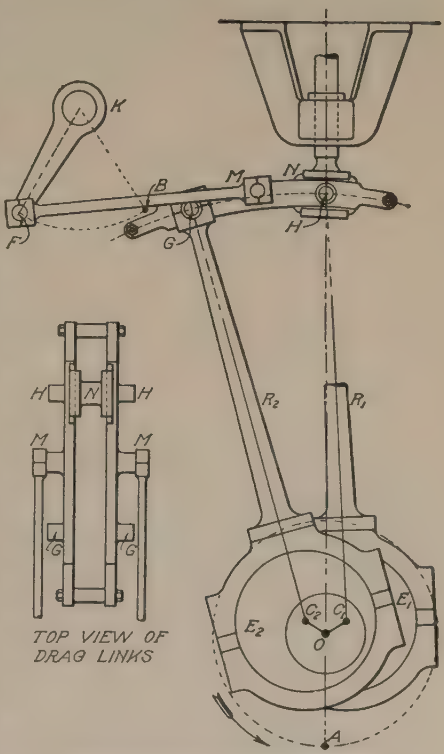

Fig. 52. Stephenson Link Motion

The valve stem has, at its lower end, a pivoted block N, called the *link block*, provided with slotted sides through which the links

can slide. The reverse shaft, or rock shaft, K, here shown in the full gear "forward," may be turned until F moves to B; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R_2 instead of from R_1, as before. The link in this position would be in full gear "backward."

From the foregoing, it is obvious that the Stephenson gear may be divided into three distinct mechanisms, each of which perform a definite function. *First,* the link motion proper, comprising the parts from the axle to the link; *second,* the adjusting gear, which is composed of the lifting shaft and reversing lever by which the power of the engine is controlled by lowering or raising the link; and, *third,* the valve and its attachments. The link motion proper is,

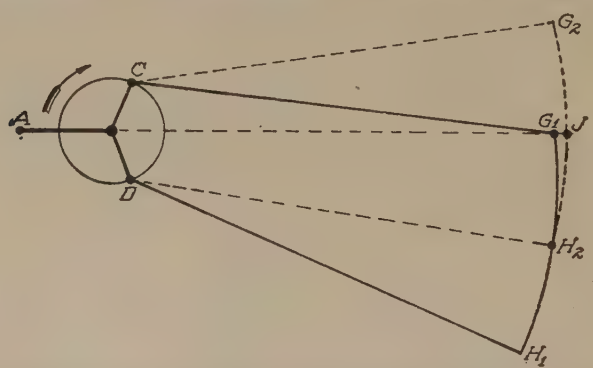

Fig. 53. "Open Rod" Arrangement of Eccentric Rods in Stephenson Gear

perhaps, the most important of the three, at least for the present study. Remembering that the link supplanted the gab-hook, it should be obvious that the eccentric rods and their connection to the link form a combination similar in action to the gab-hooks and valve rod, with some intervening parts which do not materially affect or change the operation.

Relative Position of Eccentric Rods. In order to have clearly in mind just what action does take place when the link is shifted from one position to another, it is essential that the relative position of the eccentric rods be understood. They are designated as "open rods" when arranged as shown in Fig. 53, with the eccentric centers C and D on the same side of the axle as the link, and "crossed rods" when the rods cross as illustrated in Fig. 54. The location, length, and attachment of the eccentric rods to the link have a mate-

rial effect upon the movement of the valve. Experience and calculations have shown that the eccentric rods should not be shorter than eight times the throw of the eccentric. They are usually much longer than this. The distance between the eccentric rod pins should not be less than two and one-half times the throw of the eccentric. If the distance is less than this amount, the angle between the link and the block will be such that there will be an excessive slip of the block and undue stresses in the mechanism will be induced. The angularity of the eccentric rods produces irregularities in the movement of the valve, which can be largely compensated for by locating the saddle pin inside the center line of the arc, but not too far inside for then it would give a long slip of the link and be objectionable. The adjustment of the link also requires that special attention be given to the amount of lead at full gear as well as to the increase of lead produced by "hooking up" the engine. With "open rods" it will be seen that when in full gear the link block is at G_1, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 53, and the valve must consequently be moved toward the right an amount equal to G_1J, thereby increasing the lead on the crank end of the cylinder. With "crossed rods," moving the link from full to mid gear moves the link block from G_1 to J, Fig. 54, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods, the greater this increase or decrease will be.

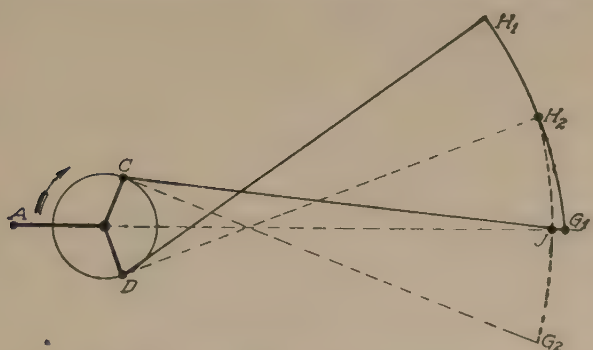

Fig. 54. "Crossed Rod" Arrangement of Eccentric Rods in Stephenson Gear

The open rods are more generally used than the crossed rods; especially is this true in locomotive service. The feature of increasing lead from full to mid gear is the distinguishing characteristic of the Stephenson gear. When the engine is in full gear, so that the forward link pin G_1 is on the center line as in Fig. 53, then only the eccentric C controls the valve, and the travel of the valve will be equal to twice the throw of the eccentric C. In other words, when in full gear, only one eccentric moves the valve, as was the case when using gab-hooks. As the link is raised, both of the eccentrics have an effect on the motion of the valve, the result being very much the same as if another controlling eccentric of shorter throw were introduced. The throw of this resultant eccentric would decrease until the center was reached, when it would be a minimum. Finally, the

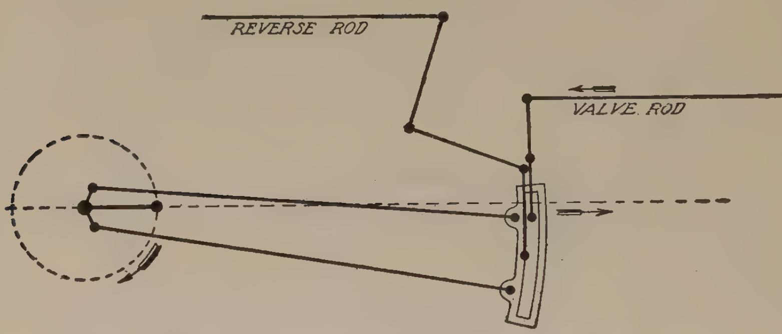

Fig. 55. Diagram of Stephenson Gear Showing Link Block and Rocker

center of this resultant eccentric would be on the center line of the motion, midway between the two actual eccentrics. At this point, the radius of the resultant eccentric would be equal to the sum of the lap and lead in full gear. Therefore, in mid gear, the valve travel is equal to twice the sum of the lap and lead in mid gear.

Location of Link Block. Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement, indicated in Fig. 55, makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 53, because the rocker reverses the direction of motion of

the valve. While apparently the crossed rod arrangement is used, yet it is really the open rod arrangement and gives increasing lead toward mid gear. A rod from the bell crank lever on the reverse shaft E leads back to the engineer's cab and connects with the reverse lever. This lever moves over a notched arc and may be held by a latch in any one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is, an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 52, the radius of the link arc is equal to C_1H and C_2G.

Exact equality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Application to Expansion and Cut-Off. Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon, the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke and the link is gradually moved toward mid gear or "hooked up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in Fig. 52, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs and, in order to give symmetrical action for both forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

Zeuner Diagram for Stephenson Gear. The Stephenson link can not be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials, a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip of the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Zeuner diagram for a Stephenson gear embodies all of the principles of the Zeuner diagram for a simple valve, with certain additional ones which, while comparatively simple, sometimes cause confusion. It is only necessary to remember that there are two eccentrics and that their combined action is the same as one virtual eccentric; also, that in passing from a long to a short cut-off with open rods, the lead increases, hence the path of the virtual eccentric center must be a curved one. A practical example will make the construction of such a diagram clear.

Example. Given a maximum valve travel of $5\frac{1}{2}$ inches, steam lap 1 inch, lead at full gear $\frac{1}{16}$ inch, and $\frac{R}{L}$ equal to $\frac{1}{4}$. Find the valve travel for 60 and 80 per cent cut-off, respectively.

Solution. Construct the valve travel circle $A\ B\ C\ D$, Fig. 56, having a diameter of $5\frac{1}{2}$ inches. (The scale of the drawing is exactly $\frac{3}{4}$ size.) Draw the lap circle $T\ U\ V\ W$ and lay off the full gear lead $S\ T$. Lay off the angles $P\ O\ B$ and $Q\ O\ D$ equal to the angle through which the eccentrics must be turned in order to displace the valve by an amount equal to the lap plus the lead at full gear; or with slight error draw a perpendicular to $A\ C$ through

the point S and where it cuts the maximum valve circle, as at P and Q, will be the centers of the eccentrics sought. Two points of the locus of the virtual eccentric center have thus been established. In order to draw the locus, the amount of lead at mid gear must be known. By the construction of Fig. 53, it was shown that in shifting the link from the full gear position G_1H_1 to the mid gear position G_2H_2, the lead was increased by the amount G_1J, which can be measured directly from the drawing. In this problem assume that the

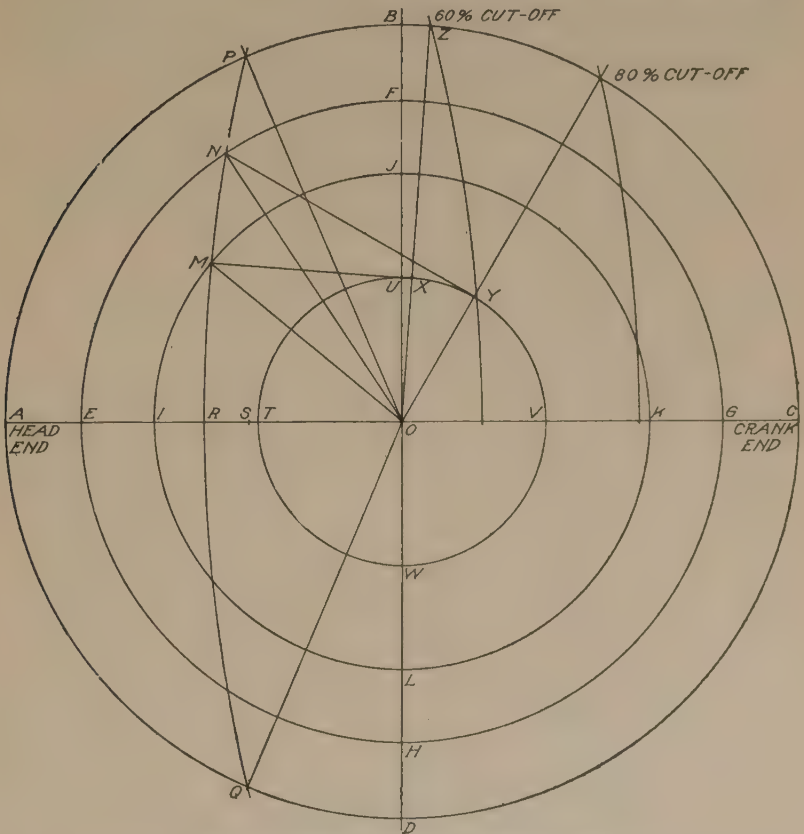

Fig. 56. Zeuner Diagram for Stephenson Gear

length of the eccentric blades is known and that by a construction similar to that in Fig. 53, the lead at mid gear was found to be $\frac{3}{8}$ inch, or, in other words, in passing from full to mid gear the lead was increased $\frac{5}{16}$ inch. Knowing the lead at mid gear, lay off the distance TR equal to $\frac{3}{8}$ inch. The locus of the virtual eccentric center must pass through P, Q, and R and have its center on the line $A\ C$ extended. By trial, we find such a center and such a radius that the arc when drawn will pass through the points P, Q, and R. This arc is the locus of the virtual eccentric center when dealing with the head-end events. To find the events for the crank end, construct a similar arc on the right of the

vertical line $B D$. To obtain the valve travel for 60 per cent cut-off, first determine the crank position in the usual manner by locating the line $O Z$, remembering that $\frac{R}{L} = \frac{1}{4}$. Where this line cuts the lap circle, as at X, draw a tangent to the lap circle and extend it until it cuts the arc $P R Q$ at M. $O M$ is then the radius of the valve travel circle for 60 per cent cut-off. Construct the valve travel circle $I J K L$ with a diameter of $3\frac{1}{2}$ inches, the required valve travel. In like manner, establish the point N, which determines the valve travel circle $E F G H$ for 80 per cent cut-off, the diameter of which is $4\frac{1}{2}$ inches. By this same procedure, the valve travel for any cut-off

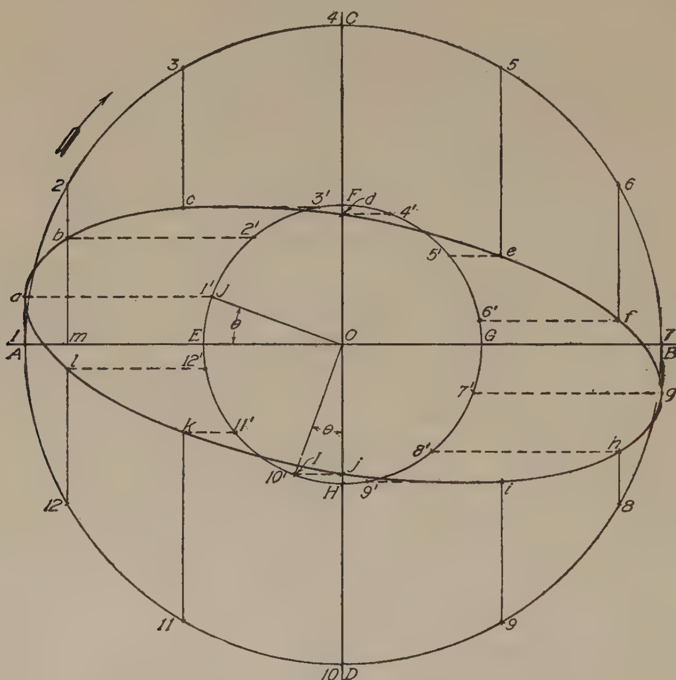

Fig. 57. Valve Ellipse Diagram for Studying Valve Action

may be obtained. Having the valve travel circle established, all the events of the stroke may be found as has already been pointed out in the study of the Zeuner diagram.

Valve Ellipse Diagram. The valve ellipse diagram furnishes another method for studying the valve action, aside from that furnished by the Zeuner diagram. The valve ellipse has been used a number of years as a means for representing the relative positions of the valve and the piston.

The principle of its construction as applied to the arrangement of valve and rods, as shown in Fig. 55, is to draw lines at right angles to each other, one representing the travel of the piston, the other that of the valve. Thus, in Fig. 57, draw the circle *A B C D*, having a diameter equal to the stroke of the piston drawn to a predetermined scale. This circle represents the path of the crank pin center. Divide this circle into any number of equal divisions, in this case, twelve, at points *1*, *2*, *3*, etc. It is evident that if a line be drawn from any one of these points, as *2*, perpendicular to the line *A B*, that, neglecting the angularity of the connecting rod, the distance *A m* would represent the displacement of the piston as the crank moved forward from *A*. To allow for the angularity of the connecting rod, take a radius equal to the length of the connecting rod drawn

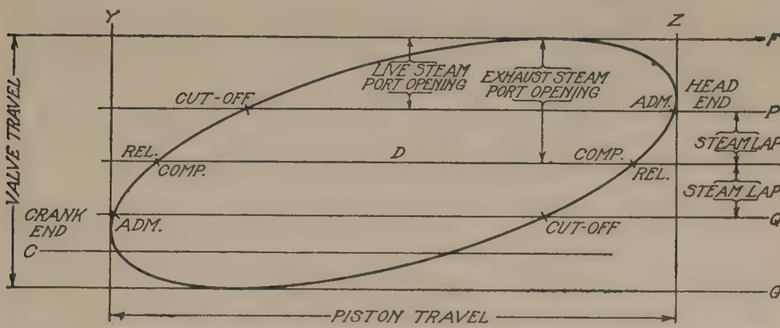

Fig. 58. Valve Ellipse Diagram Showing Information to be Obtained from its Analysis

to the same scale as that of the circle *A B C D* and sweep the arcs from the points *1*, *2*, *3*, etc., with the center on the line *A B* produced. Now representing the path of motion of the valve by the line *H F*, drawn perpendicular to *A B*, the eccentric position *O I*—which is located at the angle $(90-\theta)$ degrees behind the crank—is, for the sake of convenience in constructing the ellipse, located at *O J*. Having drawn the valve travel circle *E F G H*, begin at *J* and lay it off into the same number of equal parts as was done in the case of the crank circle. For the crank position *A*, the corresponding eccentric position is *J*, and hence, by projecting a vertical line and a horizontal line from *A* and *J*, respectively, the point *a* is located. In the same manner, the points *b*, *c*, *d*, etc., are located, thus completing the construction of the ellipse. The ellipse may have

different inclinations to the reference line, depending on conditions. This difference will be noted in comparing Figs. 57 and 58.

Thus far the discussion has dealt only with the construction of the ellipse. It is now proposed to point out what information may be obtained from the valve ellipse and for the sake of clearness, another figure is shown. After constructing the ellipse or having obtained it directly by an instrument specially constructed for the purpose, draw the reference line C in Fig. 58. Tangent to the ellipse, draw the lines F and G parallel to C. The distance between the lines F and G represents the travel of the valve. Midway between F and G draw the line D, the center line of the extreme travel of the valve. Assume that the valve is an ordinary plain slide valve having $1\frac{1}{8}$ inches steam lap and the zero exhaust lap, or line and line. Draw the lines P and Q $1\frac{1}{8}$ inches on each side of the center line D, and where these lines cut the ellipse determines the points where admission and cut-off occur for the two ends of the cylinder, as indicated in Fig. 58. Since there is no exhaust lap, the point where the line D cuts the ellipse gives compression and release for the two ends of the cylinder. In this case, the compression occurs on the head end at the same time that release occurs on the crank end, and *vice versa*. If the valve be given exhaust lap, it would be laid off in the same manner as the steam lap. Draw the lines Y and Z tangent to the ellipse and perpendicular to the reference line C. The distance between these two lines represents the length of the stroke of the engine drawn to scale. To find the per cent of the stroke at which any event occurs, it would be only necessary to drop a perpendicular to the center line D from the point on the ellipse corresponding to the event under consideration and obtain the percentage as previously explained. If the width of the steam port be known, it would be laid off from the lines P and Q toward the lines F and G as indicated. Assuming admission on the head end to occur as marked on the line P, it is evident from the portion of the curve contained between the lines F and P that at the beginning of the stroke the steam port was opened rather quickly and that cut-off occurred by the port being closed very slowly. During this time, the piston moved approximately three-quarters of the stroke. There being no exhaust lap or inside clearance, release occurred when the valve reached its central position. At the same

time that head-end release took place, compression began on the crank end, then followed crank-end admission, cut-off, release, and head-end compression, in regular order.

The valve ellipse has been largely used by steam railroad engineers and, as a result of the demand for such information as can be obtained from a consistent study of it, several devices have been invented for taking the ellipse directly from the engine. These devices consist of a drum the circumference of which is made proportional to the stroke of the engine. A sheet of paper is held on this drum by means of clips somewhat in the same manner as are found on the drums of steam engine indicators. This drum is mounted on a frame and when in use is placed in a convenient position above the crosshead or on the steam chest in such a position that its axis of rotation is perpendicular to the direction of the motion of the valve. The drum is rotated by means of a cord connection with the crosshead. Attached to the apparatus is a pencil which receives the same motion as that of the valve by means of a connection with the valve rod. Hence by the combination of the two movements, that is, of the drum moving with the piston and that of the pencil moving with the valve, the valve ellipse diagram is drawn.

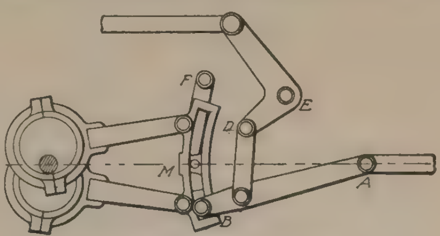

Fig. 59. Gooch Link Motion

From the study of the Stephenson gear, it is obvious that it is very flexible, and that it is readily adjusted to all irregularities of operation. Great care must be taken in its design in order that it may properly perform its work. Owing to the large number of parts and the size of same on large engines, it frequently gets out of alignment, its parts wear considerably and, on locomotives, the lubrication is sometimes difficult. On this account, it requires frequent attention in order that the best results may be obtained. All things considered, it is doubtful whether any other reversing gear gives as good a steam distribution as does the Stephenson gear when it is properly adjusted and operated.

Gooch Link. The Gooch link, illustrated in Fig. 59, has been extensively used on European locomotives, although it is gradually

being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to *A B*, the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft *E* which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin *M*, which is connected by a rod to the fixed center *F* so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to *A B*, it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point *A*, which is on the end of the valve rod. The lead then is constant for all positions of the block. The gear is more complicated than the Stephenson and requires nearly double the distance between shaft and valve stem.

RADIAL TYPE OF VALVE GEAR

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with a gear like the Stephenson if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either increased or diminished according as the link is arranged with "open rods" or "crossed rods." Moreover, the compression is hastened by bringing the link to mid gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion led to the design of the so-called "radial valve gears," many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

Hackworth Gear. The essential features of the Hackworth gear are indicated in outline in Fig. 60. In this figure *S* is the center of the shaft, and the eccentric *E* is set 180 degrees from the crank

SH. At the right-hand end of the eccentric rod EA is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its axis P to reverse the engine or to change the cut-off. P is a pivot which is located on a horizontal line through S in such a position that DP is equal to EA. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward" (as shown in the figure)

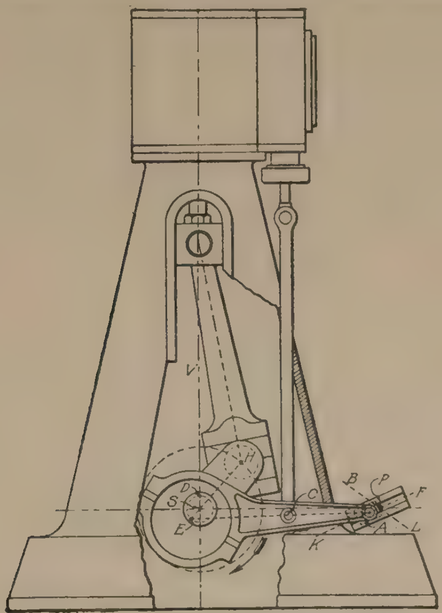

Fig. 60. Diagram of Hackworth Radial Valve Gear

through the horizontal position to "full gear back" (as shown by the line BL) without moving the valve. It will be observed, therefore, that the leads are constant for all positions of the guide. The valve rod running upward from C connects with the valve stem, which it moves in a straight line. The valve stem is made just long enough to equalize both leads and, if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear is, perhaps, the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 61, is designed to eliminate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever FP remains stationary, but can be turned on its axis P to reverse the engine or to change the point of cut-off. The pivot P is located precisely as

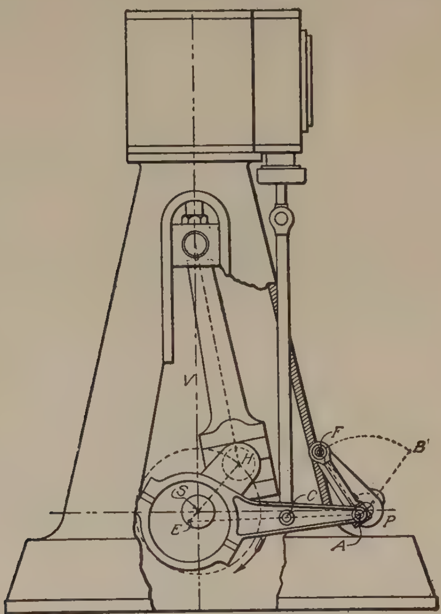

Fig. 61. Diagram of Marshall Radial Valve Gear

in the Hackworth gear, and the lever FP can be turned from "full gear forward" (as shown in the figure) to "full gear back" (as shown by the line BP), intermediate positions giving different cut-offs the same as with the Hackworth gear. Since FA is made equal to FP, the point A will always swing through P no matter where F may be and will coincide with P when the engine is on dead center. The leads for all positions of the gear, therefore, will remain constant, as in the preceding case.

The Marshall gear is sometimes made with the point C located on the right of A, on the line EA produced. In this case, if the same kind of valve is to be used, the eccentric E must move with

the crank instead of 180 degrees from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. The Joy radial gear, Fig. 62, is perhaps the most widely known, and is certainly one of the best radial gears. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from *C*, a point on the connecting rod. *H* is a fixed pivot supported on the cylinder casting. The lever *E D* has a block pivoted at *A*, which slides back and forth in a slotted guide, having a slight curvature, the concave side being toward the right. The guide and the lever *P F* are fastened to the reverse shaft *P* and, by means of a reverse rod leading off from *F*, can be turned from full gear forward, as shown, to full gear back, when the pin *F* moves over to the position *B*. Motion is transmitted to the valve stem by means of the radius rod *E G*.

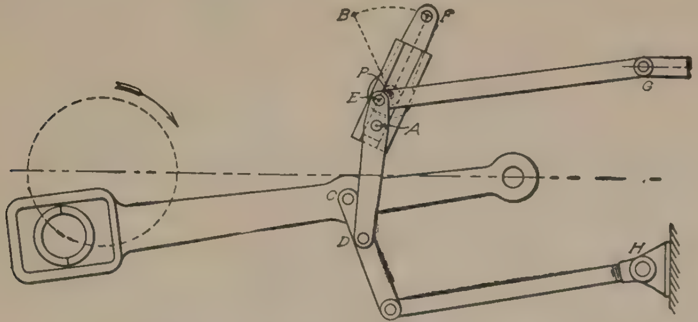

Fig. 62. Diagram of Joy Radial Valve Gear

The proportions are such that when the crank is on either dead point, the pivot of block *A* coincides with *P*, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all positions of the gear. Its many joints cause wear and its position near the crosshead makes a careful inspection of the crosshead and piston rather difficult while the engine is running.

Walschaert Gear. The Walschaert gear, Fig. 63, stands today as the best representative of the radial gear type. It has for many

years been very largely used on all the important European railroads. It has been used considerably in England and at the present time is being applied to a great many locomotives in America.

Analysis of Valve Motion. When the Walschaert gear is used, the valve receives its motion from two distinct sources, namely, from the crosshead and from the eccentric crank. In Fig. 64 the various parts of the gear are named. The crosshead connection gives to the valve a movement equal to the lap plus the lead, at the extremities of the stroke, when the eccentric crank is in its mid-position. The eccentric crank leads the main crank by an angle of 90 degrees for a valve having external admission, and follows the

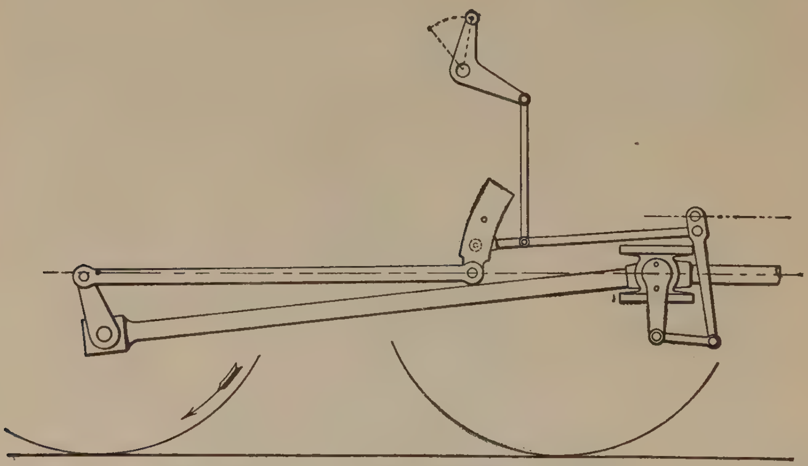

Fig. 63. Diagram of Walschaert Radial Valve Gear

main crank by 90 degrees for an internal admission valve. Locating the eccentric crank exactly 90 degrees in advance or behind the main crank is one of the necessary adjustments of the Walschaert gear. It is evident, therefore, that if an eccentric rod of proper length be attached to the eccentric crank and the valve through proper means, when the engine is on dead center, the valve would be in mid-position. However, in order to have economic operation, it becomes necessary to have some lead at dead center positions, hence the valve must be displaced by an amount equal to the lap plus the lead. Since the eccentric crank must be 90 degrees from the main crank, some other means must be used to

obtain the proper displacement and the method of accomplishing this on the Walschaert gear is one of its most distinguishing features. An attachment is made between the crosshead and the valve stem by means of a lever known as the combination lever, or, as shown in Fig. 64, the lap and lead lever.

In order to obtain the proper displacement of the valve when the engine is on dead center, the attachment of the combination lever to the crosshead and to the valve rod must bear a definite ratio to the stroke and valve travel. In other words

$$S:t \text{ as } L:V$$

or

$$V=\frac{L\,t}{S}$$

in which S is stroke of piston in inches; t is twice the sum of the lap plus the lead in inches; L is distance between the crosshead connec-

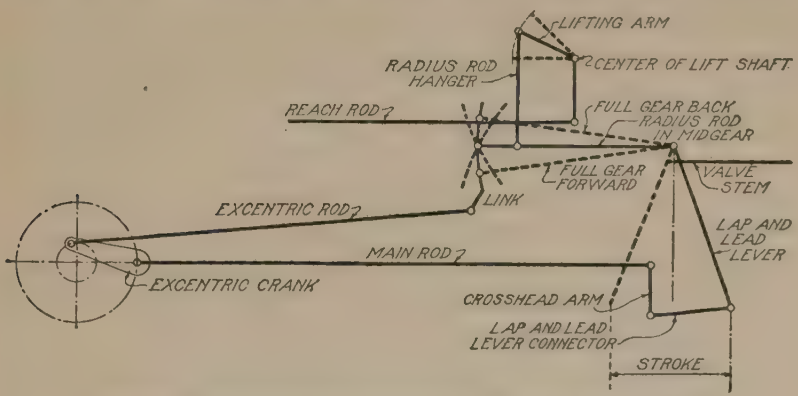

Fig. 64. Diagrammatic Analysis of Walschaert Valve Gear Action

tion and that of the radius lever in inches; and V is distance between the connection of the radius lever and that of the valve stem in inches. The above expression holds good for either an inside or an outside admission valve.

When using an inside admission valve, the connection between the radius rod and the combination lever is made above the valve stem connection as shown in Fig. 64, that shown in Fig. 63 being the arrangement for an external admission valve.

Link Motion. We have thus far traced the movement of the valve, taking into consideration the crosshead and eccentric crank connection and omitting for the sake of clearness the link connection. It will be noted that the link is pivoted at the center. The link block is raised or lowered by means of the reverse lever and bell crank. The link block is connected to the radius rod, which has a length equal to that of the link; hence, when the engine is on either dead center, the link block can be raised from one extreme position to the other without moving the valve. Therefore this gear, if properly constructed, gives a constant lead for all positions of the reverse lever. The proper construction, suspension, and attachment of the link to its allied parts is a very important matter and one rather difficult to accomplish. The proper location of the attachment of the link to the eccentric rod gives the designer a great deal of trouble, in obtaining the desired action of the valve. In locating the longitudinal position of the link fulcrum, consideration must be given to the length of the eccentric rod, which should have a minimum length of three and one-half times the throw of the eccentric and should be made as long as the existing conditions will permit. It should be so located that the radius and eccentric rods are approximately of equal length. The point of connection between the link and the eccentric rod should be as near the center line of motion of the connecting rod as possible, making due allowance for the angularity of the rods. To accomplish this, it often happens that the throw would be excessive. In such cases, a compromise is necessary, the point of connection being raised above the center line of motion as the case demands. It has been found in designing this gear that these considerations require shifting the eccentric crank from one to two degrees, thus making the angle between the main crank and the eccentric crank 91 degrees or 92 degrees instead of 90 degrees, as theoretically required. The angle being increased by such a small amount does not affect the movement of the valve to any appreciable extent.

Adjustment of Gear. From the foregoing brief remarks, it is to be noted that in order to secure the best results, the design of the Walschaert gear requires very accurate work. No hard and fast rules can be laid down as how to secure the best design, for each case presents different problems. The best way to secure required results is to try out the design on a model.

The American Locomotive Company gives the following suggestions for adjusting the Walschaert valve gear:

(1) The motion must be adjusted with the crank on the dead center by lengthening or shortening the eccentric rod until the link takes such a position as to impart no motion to the valve when the link block is moved from its extreme forward to its extreme backward position. Before these changes in the eccentric are resorted to, the length of the valve stem should be examined as it may be of advantage to plane off or line under the foot of the link support which might correct the length of both rods, or at least only one of these should need be changed.

(2) The difference between the two positions of the valve on the forward and back centers is the lead and lap doubled and can not be changed except by changing the leverage of the combination lever.

(3) A given lead determines the lap or a given lap determines the lead, and it must be divided for both ends as desired by lengthening or shortening the valve spindle.

(4) Within certain limits, this adjustment may be made by shortening or lengthening the radius bar but it is desirable to keep the length of this bar equal to the radius of the link in order to meet the requirements of the first condition.

(5) The lead may be increased by reducing the lap, and the cut-off point will then be slightly advanced. Increasing the lap introduces the opposite effect on the cut-off. With good judgment, these qualities may be varied to offset other irregularities inherent in transforming rotary into lineal motion.

(6) Slight variations may be made in the cut-off points as covered by the preceding paragaph but an independent adjustment can not be made except by shifting the location of the suspension point which is preferably determined by a model.

Zeuner Diagram for Walschaert Gear. The Walschaert gear may be examined by the aid of a Zeuner diagram to the same limited extent as the Stephenson. The construction of the Zeuner for a Walschaert gear is somewhat easier than for the Stephenson because the locus of the virtual eccentric centers lie on a straight line, due to the constant lead. For example, take a maximum valve travel of $5\frac{1}{2}$ inches, a lap of 1 inch, and a lead of $\frac{3}{16}$ inch. In Fig. 65 (scale of drawing is exactly three-fourths size), the valve travel circle is $ABCD$, the lap circle $ILMN$. Lay off the given lead IF $\frac{3}{16}$ inch and through the point F erect a perpendicular line cutting the circle $ABCD$ at H and E, thus locating the two eccentric positions. Since there is a constant lead for any valve travel, the line HFE becomes the locus of the virtual eccentric centers. Assuming a cut-off of 80 per cent, locate the line OK and at J, the point where this line cuts the steam-lap circle, erect a perpendicular line

and extend it until it cuts the line $H\,F\,E$ at G. The point G is the extremity of the valve travel circle for 80 per cent cut-off, the radius

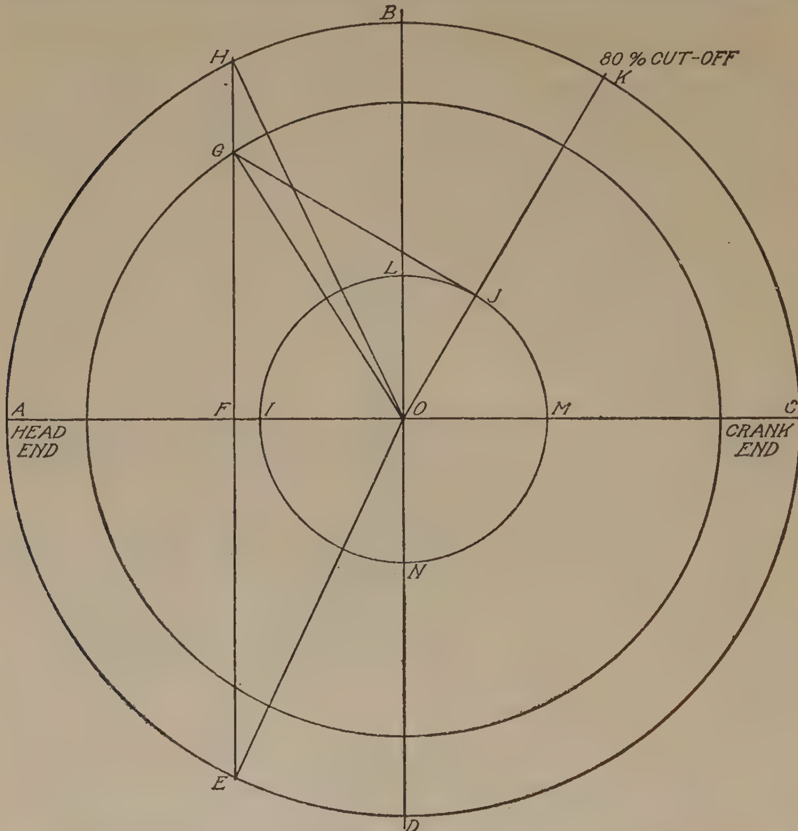

Fig. 65. Zeuner Diagram for Walschaert Gear

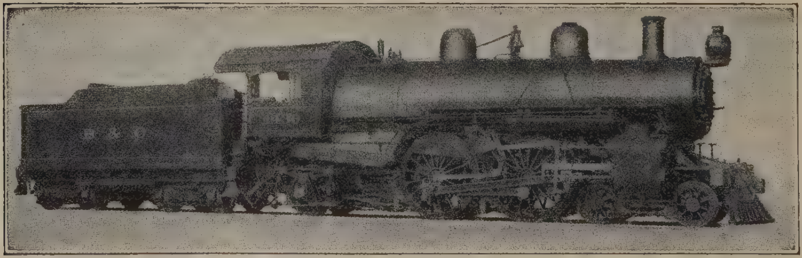

Fig. 66. Application of Walschaert Valve Gear to a Passenger Locomotive

of which will be $O\,G$, $2\frac{3}{16}$ inches. In like manner, the valve travel can be obtained for any other point of cut-off.

Dimensions of Walschaert Gear Parts. For an engine such as is shown in Fig. 66, an approximate value of the various rods and levers may be taken as follows. By referring to Fig. 64 the location of the various parts can be determined more readily than in Fig. 66.

Main rod	8′–0″	Radius rod	3′–10″
Eccentric crank	1′–2″	Lap and lead lever (total)	3′– 0″
Eccentricity	$6\frac{1}{2}$″	Lap and lead lever connector	1′– 2″
Eccentric rod	4′–6″	Crosshead arm	1′– 0″
Link arc	1′–10″	Stroke	2′– 0″

DOUBLE VALVE GEARS

It has been shown that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at a short cut-off owing to excessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear; and also by the use of two valves instead of one.

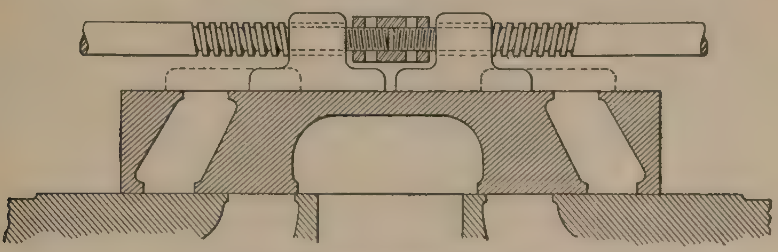

Fig. 67. Section of Meyer Double Valve

The main valve controls admission, release, and compression; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve, sometimes known as the riding cut-off valve, may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer valve, Fig. 67. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right-hand and a left-hand thread, so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve spindle between the

steam chest and the head of the valve rod, and the valve spindle is prolonged into a tail rod which projects through a stuffing box on the head of the steam chest, Fig. 68. The end of this tail rod is square in section and reciprocates through a small hand wheel, by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric which moves the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off and this will also give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction opposite to that of the main valve. The valve may be designed by means of the Zeuner diagram.

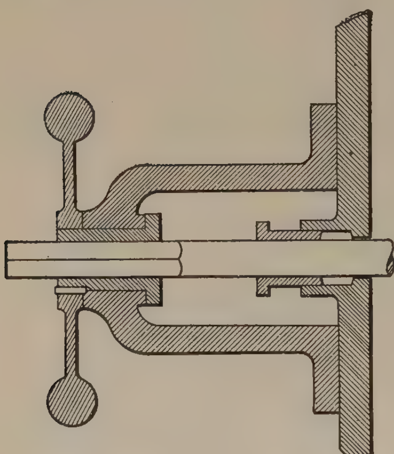

Fig. 68. Stuffing Box and Valve Spindle of Meyer Gear

Design by Zeuner Diagram. Let us design a Meyer valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 5 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be 1 to 6.

In Fig. 69, draw $X\ O\ Y$, the main valve travel, equal to 4 inches. Lay off $X\ D$ equal to 5 per cent of 4 or 0.2 inches, and with a radius of 12 inches, and the center on $Y\ X$ produced, draw the arc $D\ H\ K$. $H\ K\ O$ is the crank position at compression on the head end; $C\ K\ O$, the crank position at compression on the crank end, is found in a

similar manner. Lay off $O\ I$ equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this, take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B. B is the center of the valve circle and $O\ B\ E$ is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this $1\frac{1}{2}$ inches) draw an arc.

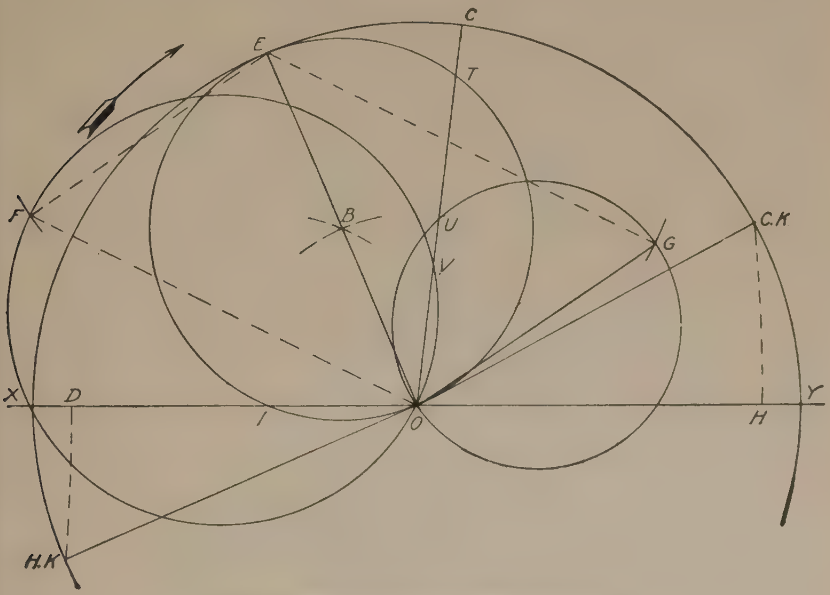

Fig. 69. Zeuner Diagram for Meyer Valve Gear

With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F. On $O\ F$ as a diameter, construct a valve circle. This valve circle will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to $F\ O$, draw an arc, and with O as a center and with a radius equal to $E\ F$, draw another arc intersecting the first at G. On $O\ G$ as a diameter, construct a valve circle. This circle will then represent the travel of the cut-off valve moving on the main valve. That is, it will represent the

displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking, a valve circle, and $O\ G$ is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

Draw the crank line $O\ C$ at any position, cutting the valve circles at T and U and V. $O\ V$ represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest, and $O\ T$ represents the displacement of the main valve. The relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between $O\ V$ and $O\ T$, since both valves are moving in the same direction. By careful measurement it will be found that $O\ U = O\ T - O\ V$, and any arc as $O\ U$ on the auxiliary circle $O\ U\ G$ will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

In Fig. 70 are shown $H\ K$ the crank angle at head-end compression, $C\ K$ the crank angle at crank-end compression, the main valve circle, and the auxiliary circle, all of which have been transferred from Fig. 69. In order to avoid confusion, the construction lines and all lines not essential to the figure are omitted.

Lay off on Fig. 70, $O\ I$ equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle $H\ E\ O$. It will intersect the valve circle for the main valve at L and M. Draw $H\ O$ through L, representing the crank position at admission (head end) and $O\ M\ \ H$ through M showing the crank position at cut-off. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course, a later setting would be of no avail, for the port would be closed by the main valve at this angle. The crank line $O\ M\ \ H$ cuts the auxiliary circle at N_1 so that $O\ \ N_1$ ($1\frac{15}{32}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{15}{32}$ inches from the edge of the main valve port in order to cut off at this crank angle. The full lines of Fig. 66 show the cut-off valve placed in this position.

The intersection of $H\ K\ O$ with the lower valve circle gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to $H\ \ K\ O$ from E_2. This cuts the

circle at P, and $O\ P$ equals the head-end inside lap $\frac{1}{32}$ inch, and $H\ E\ I$ represents the corresponding lap circle.

The crank position at compression on the crank end is $C\ K$, which cuts the upper valve circle at N_2, giving an inside lap for the

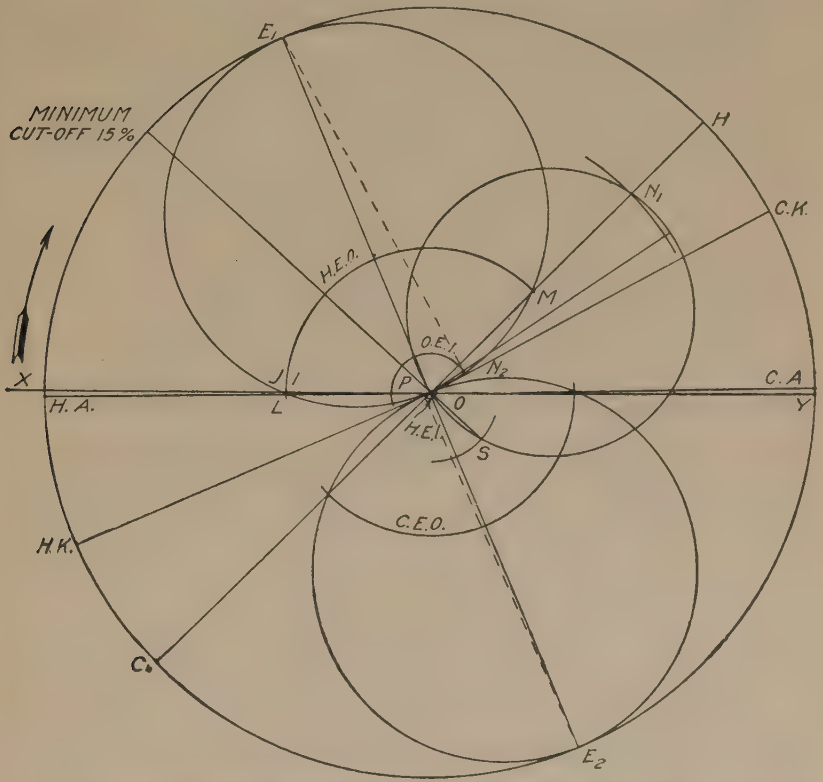

Fig. 70. Zeuner Diagram Showing Additional Factors in Design of Meyer Valve

crank end, of $O\ N_2$, $\frac{13}{64}$ inch. To make this intersection more apparent, the perpendicular may be drawn from E_1, as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression. Produce this line through O until it cuts the auxiliary circle at S. Then $O\ S$ equals the required lap $\frac{23}{64}$ inch for the cut-off valve in order to cut off at 15 per cent of the stroke. The dotted lines in Fig. 67 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be $1\frac{1}{2}$ inches wide and the valve port 1 inch wide. Fig. 67 shows this valve laid out to scale, but as this process is in all respects similar to that described for laying out a plain slide valve, it will not be described in detail.

Shifting Eccentric Valve Gear. In addition to the valve gears already described, there is another class which receives a very large application, particularly in small and medium-power high-speed engines. This class of gear is what may be termed the shifting

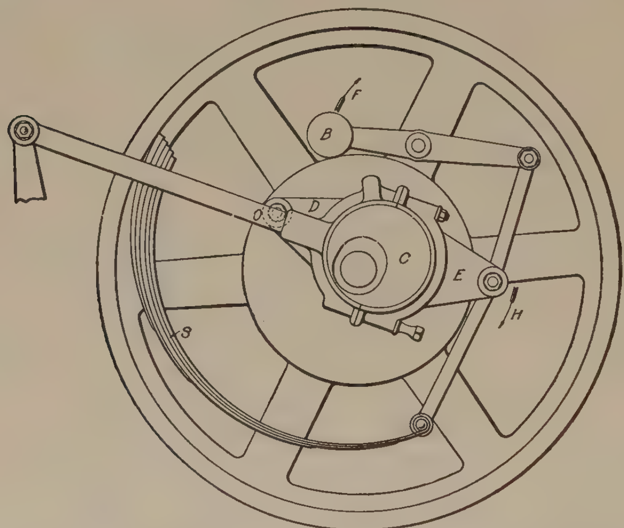

Fig. 71. Diagram Showing Action of Straight-Line Type of Shaft Governor

eccentric gear. The valve itself is the ordinary flat slide or piston valve, and the valve stem, eccentric rod, and eccentric are the same as used in the common arrangement. The difference between the fixed and the shifting eccentric lies in the method of attaching the latter to the shaft and in the mechanism provided to move this eccentric from one position to another across the shaft. The general arrangement is illustrated in Fig. 71, which represents that used on the straight-line engine.

In this, *O* is the fixed pivot of the eccentric lever *E*, and *C* is the eccentric. The pin *H* of the eccentric lever is connected through a link to a leaf spring and through the other to a weighted lever *B*, as shown. When the engine is running, the position of the weight *B*

changes under different speeds and loads, and this change in position is transmitted to the eccentric. Since O is a fixed pivot, any motion of the eccentric lever E or eccentric C must be around O as a center. Consequently, when the eccentric position changes, its center will move in a path which is an arc of a circle with a center at O. The slot

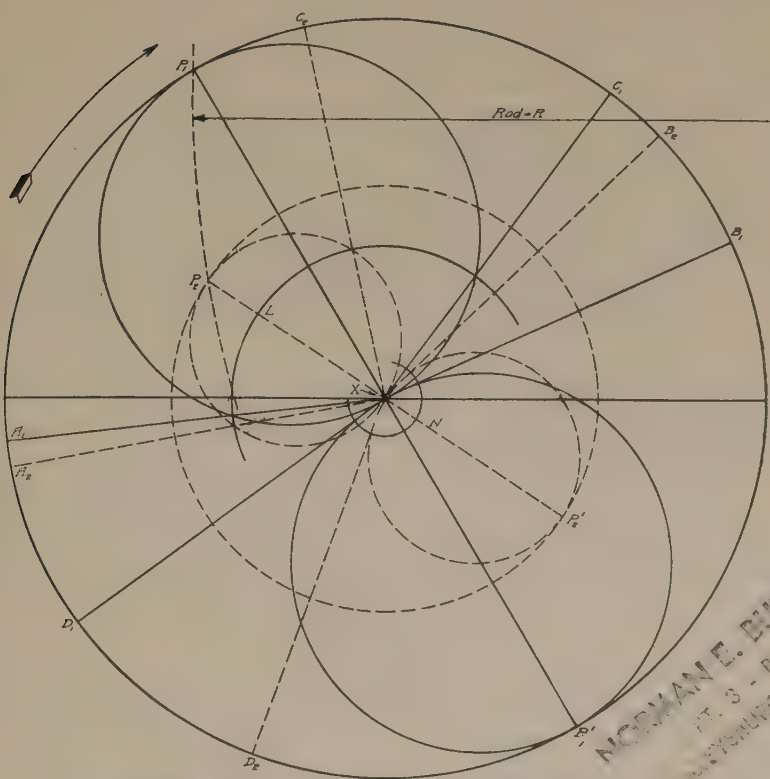

Fig. 72. Zeuner Diagram for Shifting Eccentric Valve Gear

in the eccentric is provided so as to enable it to move across the shaft to whatever position may be desired.

Two things are to be noticed for different eccentric positions. The first is that the eccentricity becomes less as the eccentric center moves along its path toward the shaft. The second is that the angular advance increases under this same condition. The first effect is to decrease the valve travel and the second to increase the angular

advance. The effect of the shifting of the eccentric on the motion of the valve is a combination of these two changes.

Fig. 30 shows the effect of changing the angular advance, and Fig. 31 shows the effect of changing the valve travel. The effect of both these changes acting together is shown in Fig. 72, which is a Zeuner diagram for a shifting eccentric valve gear.

Analysis of Zeuner Diagram. In Fig. 72 the full lines represent eccentric and crank positions at the point of maximum cut-off, and the dotted lines represent their positions corresponding to some earlier cut-off. A_1 is the crank position at admission; C_1 is its position at cut-off; B_1 is its position at release; and D_1 is its position at compression. P_1 and P_2 are the corresponding positions of the eccentric center, the subscripts referring to the maximum cut-off and to the earlier cut-off, respectively. The radius R is used to draw the path of the eccentric center and has the same length as the distance from the fixed eccentric lever pin to the center of the eccentric. The steam lap in the figure is XL and the exhaust lap is XN. The construction is made for the head end only, to avoid confusion due to the large number of additional lines required for that of the crank end. These laps remain the same for all positions of the eccentric.

Each of these two diagrams is made in exactly the same way as the ordinary Zeuner diagram and, if given the necessary data, the construction of the combined diagram should give no trouble. One point to bear in mind in drawing a Zeuner for this kind of valve gear is that the eccentric center for any position of the gear will lie somewhere along the arc described with the radius R.

An inspection of Fig. 72 shows that all events occur earlier with the earlier cut-off. However, they do not all continue for the same period, as was found to be the case when the angular advance alone was changed. This is because the valve travel is changed with the angular advance. The combined effect is that admission is advanced very slightly, cut-off is advanced a considerable amount, and release and compression are each advanced a moderate amount. Since the cut-off advances a greater amount than the release, the result is a greater expansion at earlier cut-off, which is more economical, within reasonable limits, in the use of steam. Also the increase in compression up to a certain point will make the engine run more smoothly by cushioning the piston better on the dead centers. Another effect of

earlier cut-off is a decrease of lead from that at maximum cut-off. This may or may not be an advantage, depending on the engine speed, construction of steam ports, amount of clearance, etc.

Thompson Automatic Valve Gear. The Thompson automatic valve gear, commonly known as the "Buckeye", belongs to the general class of double valve gears. Its principle of action involves certain ingenious points which make its study very interesting.

Two styles of valves of this type have been developed by the Buckeye Engine Company, namely, a flat valve and a round or piston

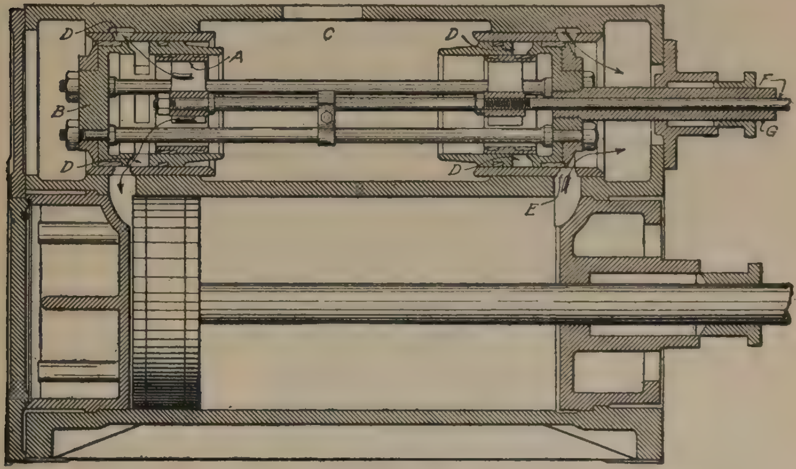

Fig. 73. Section of Cylinder and Valve-Gear Mechanism of Thompson Automatic Valve Gear

valve. While the essential features of the two valves are the same, the piston valve, illustrated in Fig. 73, is the simpler of the two and represents the latest practice. The cut-off valve *A* moves inside of the main valve *B*, and live steam entering at *C* passes through the cut-off ports *D D* in the main valve and is admitted to the cylinder, as these ports are alternately brought into coincidence with the cylinder ports *E E*, as shown by the arrows on the left. The exhaust steam is discharged at the ends of the main valve and does not come in contact with the valve except at the ends. It will be noted that the construction is such that the valve is at all times balanced.

The valve stem *F* of the cut-off valve passes through the hollow stem *G* of the main valve. Packing rings are used on both valves

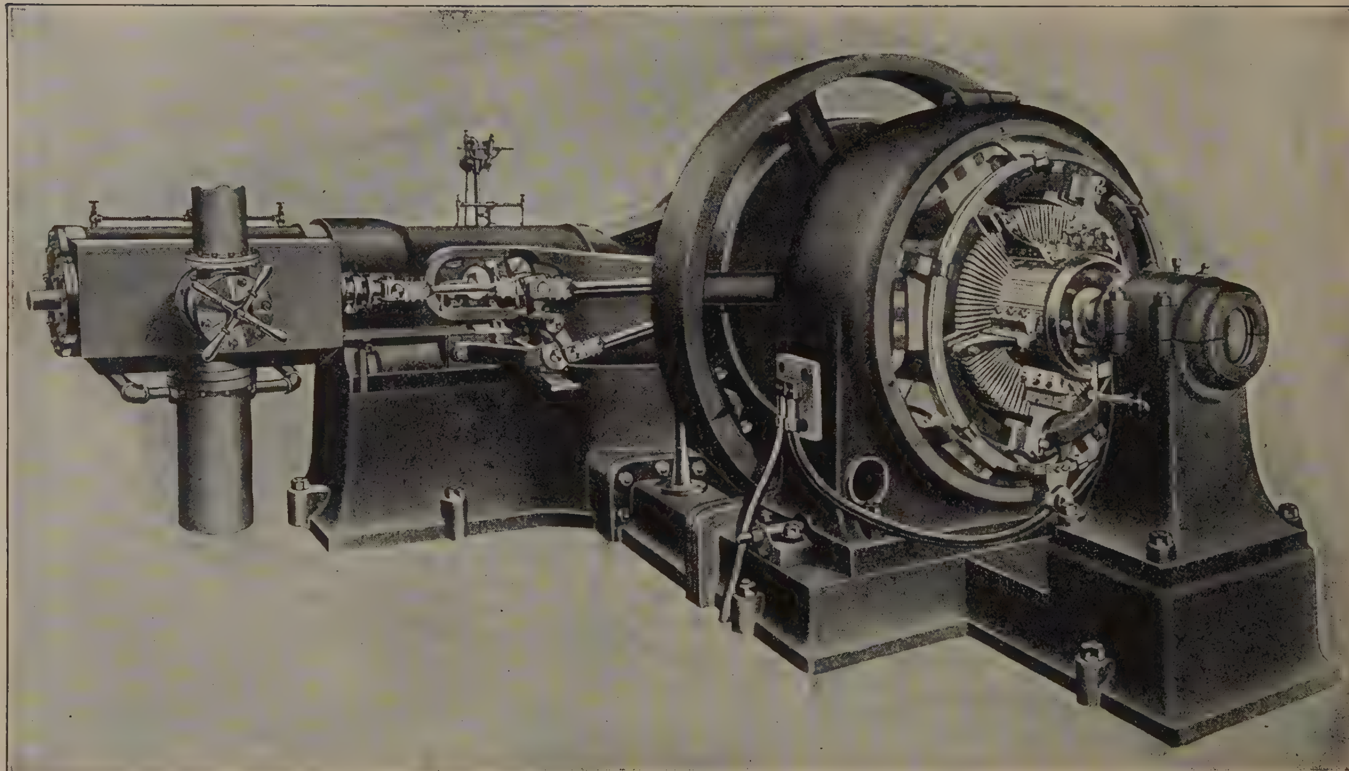

Fig. 74. Thompson Automatic Valve Gear Applied to Buckeye Engine
Buckeye Engine Company, Salem, Ohio

to insure a steam-tight connection, and bridges are provided to afford a proper bearing for the rings in passing over the ports.

Fig. 74 shows the valve gear as applied to the engine. The operation of the gear can be better understood by referring to the line drawing, Fig. 75. The crank A C is shown on the head-end dead center and running over. The eccentric D connects to the main valve stem through the eccentric rod D M, the joint M being guided by the rocker arm H M, pivoted to the engine frame at H. The cut-off valve is operated from the eccentric E by the eccentric rod E F, and the rocker arm F K N is pivoted to the rocker arm M H at

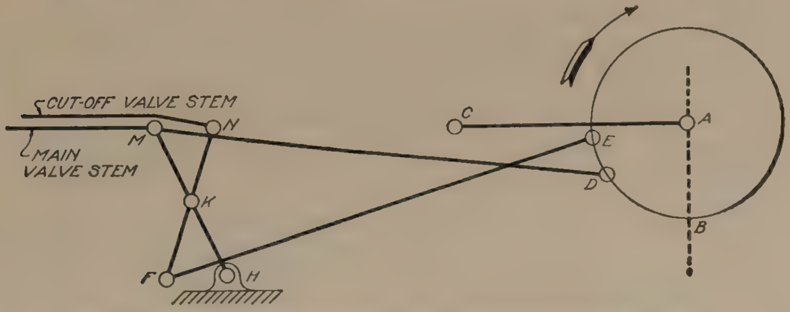

Fig. 75. Diagram Showing Operation of Thompson Automatic Valve Gear

the point K. In the compound rocker arm, the arms M K, K H, N K, and K F are all equal. On account of the valve under discussion being one having internal admission, it will be noted that the eccentric of the main valve follows the crank instead of preceding it, as is found in most cases.

Starting from the head-end dead center, suppose the crank, and likewise the eccentric, to turn through a small angle ϕ. This movement will cause the point M and the main valve to move to the left, a distance which we will call x, and the pivot point K will also move to the left a distance $\frac{x}{2}$. Now if F be considered as a fixed point, the movement of K, equal to $\frac{x}{2}$, causes the point N, and consequently the cut-off valve, to move to the left a distance x. The point F is not fixed, for while M is moving a distance x to the left, the rocker arm F K N, being pivoted to the rocker arm H K M at the point K, will cause the point F to move to the left (due to the rotation of the eccen-

tric E) a distance which we will call y. This movement of F would cause N to move to the right a distance equal to y, provided the point K were stationary. Thus it will be seen that the point N and the cut-off valve are given a movement which is the resultant of two motions and is equivalent to $x-y$; and the relative movement of the two valves would be $x-(x-y)=y$. But y is the motion which would be given the cut-off valve by the eccentric E, independent of the other mechanism; *i.e.*, the construction is such that the cut-off valve moves on the main valve in much the same manner as an

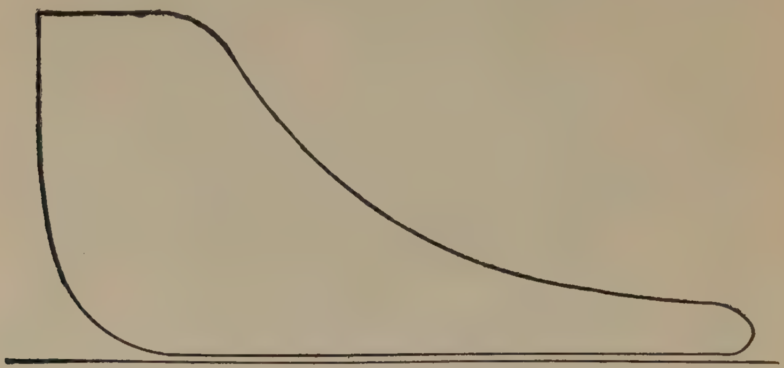

Fig. 76. Indicator Diagram, Showing Effect of Application of Thompson Gear

ordinary plain slide or piston valve moves over stationary ports when connected to a constant throw eccentric through a reversing rocker arm.

The governor in moving the cut-off eccentric simply causes it to turn about the shaft, thus cutting off the steam earlier or later, according as the eccentric is advanced or moved back on the shaft. This action takes place without changing the cut-off valve travel or the relative movement of the two valves, since the throw of the two eccentrics is equal and constant. The Zeuner diagram for the Buckeye valve is worked out in a manner similar to that described for the Meyer valve.

Two important claims are made for this valve gear:

(1) On account of the valves moving in opposite directions at the instant cut-off occurs, cut-off is made very quickly, thus eliminating quite largely wiredrawing and giving an indicator diagram having a sharp turn at the point of cut-off, resem-

bling that given by a Corliss valve gear. This is illustrated by the diagram, Fig. 76.

(2) On account of the constant travel of the valves, they wear better than those that control the regulation by varying the valve travel.

This latter claim makes the gear particularly suited for the piston valve, since uneven wear or leakage is more liable to result from the packing rings if the valve movements are variable.

DROP CUT-OFF GEARS

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve, the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two valves are placed above the center line of the cylinder and two below, the upper being for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz, a very much better distribution of steam; short steam passages and small clearances; separate ports for the admission of hot steam, and the exhaust of the same steam after expansion when its temperature has fallen; and finally the possibility of opening and closing the ports very rapidly, thus preventing wiredrawing. The small clearances, short ports, and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so, a disengaging

mechanism may be used, and the valves may shut off themselves, either by virtue of their weight or by means of springs or dashpots.

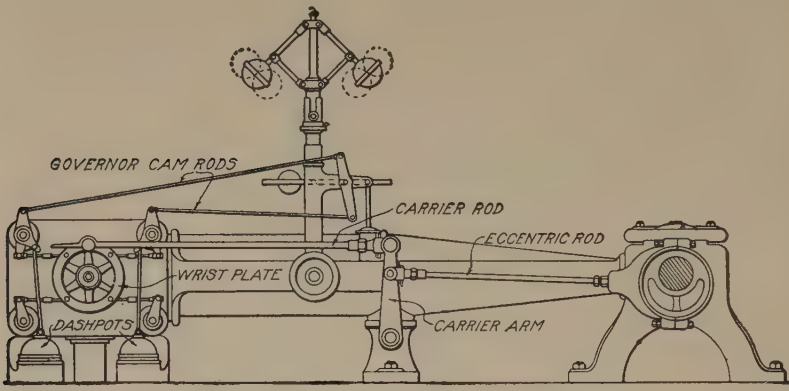

Fig. 77. Diagram of Reynolds-Corliss Drop Cut-Off Gear

The valve is usually opened by means of links or rods moved by an eccentric and, at the proper point of cut-off, the rod is disengaged from the valve, which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, Figs. 77 and 78. It is often referred to as the *Reynolds hook-releasing gear*. An eccentric on the main shaft gives an oscillating motion to a circular disk, called the *wrist*

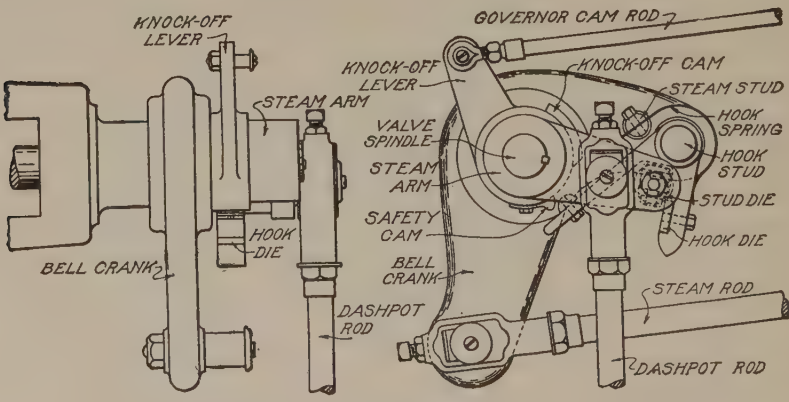

Fig. 78. Details of Reynolds Hook

plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods*

or *exhaust rods*, according to whether they move the admission or exhaust valves.

The valves which are shown in section in Fig. 79 oscillate on cylindrical seats, and the position of the rods is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 78. The steam arm is keyed to the valve spindle which passes loosely through a bracket on which the bell-crank lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the steam rod toward the right will turn the bell-crank lever and raise the hook stud. The hook (from which the gear derives its name), pivoted on this stud, has at one end a hardened steel die with sharp, square edges, and at the other end, a small steel block with a rounded face. As the hook rises, the hook die engages the stud which is fastened to the steam arm, and one end of the steam arm is thus raised. This turns the valve in its seat and admits steam. As the hook continues to rise, its stud moves in an arc above the valve spindle, and the round-faced block at its left-hand end strikes the knock-off cam, which causes the hook to turn about its stud and disengage the hook die from the stud die. In raising the steam arm, the dashpot rod is also raised and a partial vacuum is created in the dashpot. As soon, therefore, as the dies become disengaged, the dashpot quickly drops under the force of this vacuum, thus turning the steam arm and closing the valve. The striking of the left-hand end of the hook against the knock-off cam determines the point of cut-off by releasing the valve at that instant.

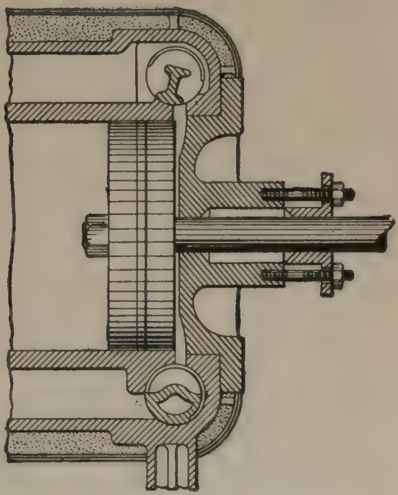

Fig. 79. Diagram, Showing Reynolds-Corliss Valves in Section

This cam is a part of the knock-off lever to which the governor cam rod is fastened. Any action of the governor which would cause

the cam rod to move toward the right would cause this knock-off lever to turn on its axis, the steam arm, and consequently lower the position of the knock-off cam. This would cause an earlier contact between the cam and the end of the hook, and consequently an earlier cut-off. By lengthening or shortening the governor cam rod, the point of cut-off can be adjusted to suit the engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that a Corliss gear operated by a single eccentric can not be arranged to cut off later than half-stroke. Suppose the eccentric is set just 90 degrees ahead of the crank. Then it will reach its extreme position just as the piston gets to half-stroke. If, by that time, the hook which was rising and opening the admission valve has not yet struck the knock-off cam, it can not strike it at all, for any further motion will cause the hook to descend to its original position, that is, its position at the beginning of the stroke; the hook will not disengage from the steam arm stud at all and the bell crank will return, closing the valve in the same manner in which it opened it. Cut-off will then take place near the end of the stroke, but it will not be the sharp cut-off produced by the sudden drop when the dies are disengaged.

If the eccentric were set less than 90 degrees ahead of the crank, the cut-off could be arranged to occur later than half-stroke, but this is decidedly impracticable, for with such a position of the eccentric, the action of the valves at release and compression is spoiled. When it is necessary to cut off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set *more* than 90 degrees ahead of the crank to operate the exhaust valves, and one *less* than 90 degrees ahead to operate the admission valves.

The safety cam, Fig. 77, is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine, and allow it to run away. The safety cam prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve consequently remains closed and the engine stops.

Nordberg Gear. The Nordberg type of drop gear is designed for high speed and hard service. Instead of having its steam arm supported on the valve spindle, it is supported on a bearing formed by an extension of the steam bonnet, and the arm is provided with two hook dies instead of one. To eliminate side strains these two dies are connected on either side of the dashpot rod. The release is accomplished by means of an extension of the steam arm which rides in a slotted cam. The position of the latter is under control of the governor. The dashpot is also carried by the steam bonnet and is located above the gear. The spring type of dashpot is used to secure positive action at high speed.

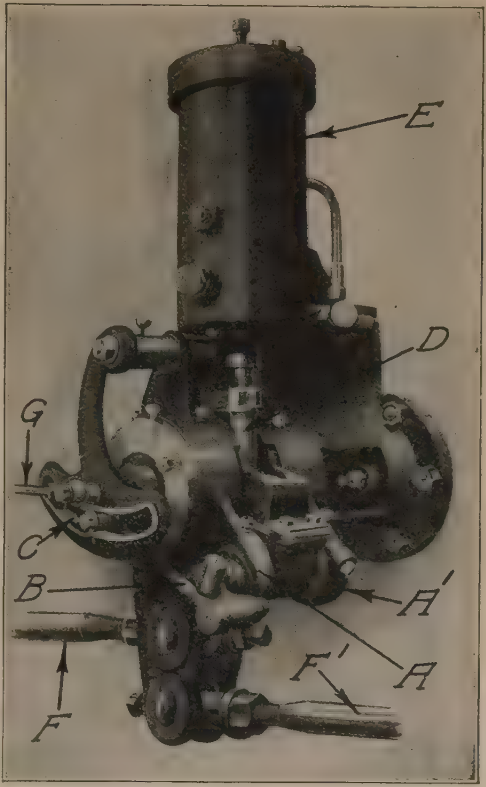

Fig. 80. Nordberg Drop Gear
Courtesy of Nordberg Manufacturing Company, Milwaukee, Wisconsin

In Fig. 80, *A A* are the two parts of the steam arm to which the hook dies (not shown) are attached. *B* is the extension of the steam arm, and it carries at its left end a roller which works in the releasing cam *C*. When this roller strikes the off-set, shown in the cam, it raises the arm *B*, which disengages the hook dies and allows the dashpot rod *D* to snap the steam valve shut. *E* is the dashpot cylinder. *F F* are the driving rods, the one on the right being connected to the eccentric on the engine shaft, and the one on the left driving the valve gear for the other end of the cylinder. The rod *G* is connected indirectly to the governor and controls the point of cut-off by changing its position as the governor changes, according to the load on the engine.

Brown Releasing Gear. In addition to the Reynolds hook, several other devices are in use for opening and releasing Corliss admission valves. Among them, the Brown releasing gear, Fig. 81, may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 78. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle; and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same place. The bearing on which it turns is carried on the lower end of the rocker and, therefore, is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

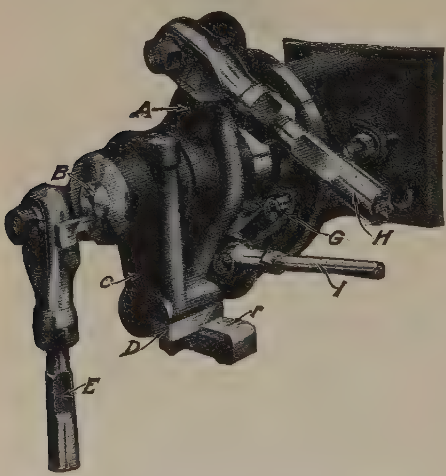

Fig. 81. Brown Releasing Gear for Operating Corliss Admission Valves

In the position shown, the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-hand direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward, the dies become disen-

gaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise, while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, Fig. 82. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to the axis of the cylinder, and the exhaust valves at right angles to the axis of the cylinder and just below it. *A A* are rock shafts turning in fixed bearings. *B B* are the admission valve stems. *C* is a slide bar, receiving a reciprocating motion from an eccentric. *T T* are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate *D*, which slides through the guide *E*. The guide *E* is made fast to the governor rod *F* and through this means can be raised or lowered, thus regulating the height of the tappets.

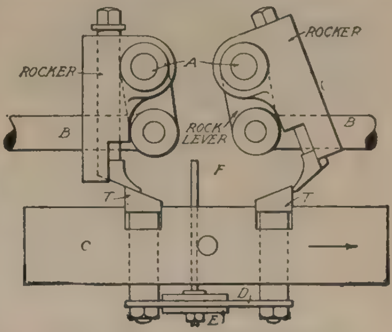

Fig. 82. Greene Drop Cut-Off Gear

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem *B* toward the right, thus opening the admission valve. Since the tappet moves in a horizontal direction, while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve, which is at once closed by a dashpot (not shown in the figure). If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and guage plate. As the slide bar *C* moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker and, both being beveled, the toe of the rocker rises in its socket, allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its return and open the valve. In this gear, the disengagement of the valve throws

no load whatever on the governor, a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

Sulzer Gear. The Sulzer gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

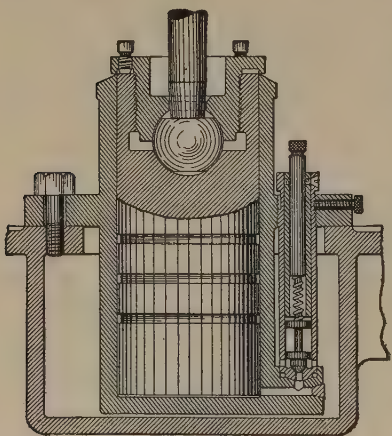

Fig. 83. Form of Vacuum Dashpot for Closing Admission Valves

The exhaust valves of all drop cut-off gears are comparatively simple in their operation, and both in opening and closing are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 83. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame, as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

CORLISS VALVE SETTING

The setting of a Corliss valve gear is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 77, the length of both the eccentric rod and the carrier rod are usually adjustable, and the former should be of such length that the carrier

arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot. Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 84, and the wrist plate should swing so that *A*, the mark it carries, moves from *C* to *D*, but not beyond either one. When *A* is in line with *B*, the wrist plate is in mid-position. The valves are then not in their exact mid-position, but it is customary to regard them as being in mid-position, and to speak of the laps as the amount which the port is covered by the valve when the wrist plate is in mid-position.

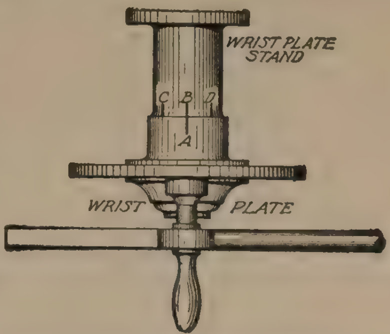

Fig. 84. Corliss Wrist Plate for Adjusting Carrier Rod

Adjusting Steam Lap. To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but on their inside the cross section is as shown in Fig. 85. On the end, in line with the finished edge of the valve and on the seat in line with the edge of the steam port, are marks, as shown in Fig. 85. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves, and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in Table II, opposite the given size of engine.

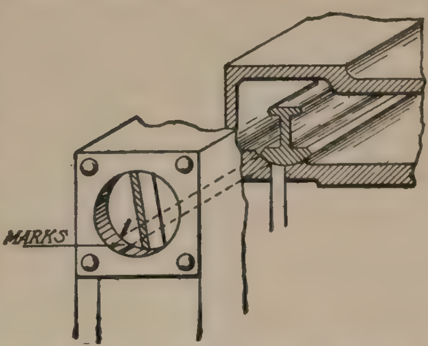

Fig. 85. Diagram Showing Method of Adjusting Steam Lap for Corliss Valves

TABLE II

Standard Lap and Clearance Values

Diameter of Cylinder Inches	Steam Lap Inches	Exhaust Clearance Inches
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
16 to 22	$\frac{3}{8}$	$\frac{1}{16}$
22 to 28	$\frac{7}{16}$	$\frac{3}{32}$
28 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Adjusting Exhaust Clearance and Lead. Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in Table II. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be $(90+\theta)$ degrees ahead of the crank. The proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}$ inch for small engines to as much as $\frac{5}{32}$ inch or $\frac{3}{16}$ inch for large engines and those of higher speed. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks, Fig. 78, descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}$ inch to $\frac{1}{16}$ inch to spare, depending on the size of the gear.

Adjusting Cut-Off. To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First, set the double lever at the right of the governor cam rods, so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor

in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook can not open the valve. A latch is provided, on which the governor can be supported slightly above its lowest position, so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up, this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

When Corliss gears are set, as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it can not be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.

VALVE GEAR TROUBLES AND REMEDIES

Importance of Keeping Valve Gear in Condition. The valve motion, or valve gear, is primarily responsible for the correct steam distribution in all steam engines. It follows, then, that in order to maintain efficient operating conditions the valve gear should receive constant careful attention in order that any irregularities which may develop can be detected at once and the fault corrected. A great many of the different gears used in American practice are described earlier in this text. In a number of cases the methods used in adjusting the gear and setting the valve have been given. For this reason the matter presented under this heading will be principally a discussion of the methods to be followed when trouble develops under conditions of service.

Familiar Types. Of the many different types of valves gears described in the preceding pages, perhaps the most familiar ones are included under the following heads:

(1) The direct-acting duplex pump valve gear
(2) The plain D-valve or piston valve gears of the simple steam engine
(3) The Corliss engine valve gear
(4) The Stephenson link motion valve gear
(5) The Walschaert radial valve gear

DUPLEX PUMP VALVE GEAR

Description. A great variety of valve gears are used in direct-acting steam pumps. The most common form, and in many respects the most reliable, is that illustrated in Fig. 50, in the text "Steam Engines". A pump such as shown in the illustration is nothing more than two pumps combined. In this particular design the motion of the piston rod of each pump is made use of in operating the valve of the other. In such a gear the only part which is made adjustable is the length of the valve rod. It is easily seen therefore that the setting of the valve is a comparatively simple matter.

Possible Troubles. After such a pump has been in service for some time, it may be necessary to dismantle it preparatory for removal to a machine shop for repairs. An accident may happen in which one of the operating arms, which are usually made of cast iron, becomes broken. In either case the operating engineer should be in a position to readjust the parts and properly set the valves after the necessary repairs have been made.

Setting Valves. The valves of such a pump are usually of the D-type, but piston valves could be used to advantage if desired. In setting the valves the general procedure should be as follows:

Preparation. Remove the steam chest or valve chest covers so that the movement of the valve relative to the ports can be measured.

Measuring Valve Travel. Move each of the pistons as far as it will go against the head in one direction and make a pencil mark on the seat of each valve at its edge, the farthest from the center of its travel. Now move the pistons against the other cylinder heads and make pencil marks on the valve seats, but on the other edge of the valves. The marks on the valve seat indicate the travel of its valve, which should be symmetrical with the ports.

Equalizing Valve Travel. If the travel is unequal, relative to the steam ports, the valve stem should be adjusted until the valve overtravels each steam port by the same amount. When the travel of each valve has been equalized, the setting may be considered finished and the valve chest covers may be replaced and parts connected.

Variation in Conditions. In the adjustments just explained it is assumed that the gear was originally proportioned properly so as to cause the valve of each pump to open early enough to

prevent the pistons from striking the cylinder heads. It sometimes happens that the closing of the valves on the water end of the pump is such as to require a slightly different setting from that explained above in order that the pistons may be reversed to prevent striking. When such is the case the necessary adjustment should be made. Each individual case will probably need different treatment and cannot be anticipated.

PLAIN SLIDE VALVE GEAR OF SIMPLE STEAM ENGINE

Types. As previously explained the plain slide valve gear of the simple steam engine is the simplest of all steam engine valve gears. On account of its simplicity it is less liable to get out of adjustment or meet with an accident which would totally disable its action. The essential elements of this gear are shown in Figs. 3 and 4, 6 to 10, 20 and 21. It is usually found constructed in one of three forms: (1) the form in which the valve receives its motion directly from the eccentric; (2) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in the same direction; and (3) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in opposite directions.

Use of Rocker Arms. The use of the rockers mentioned in the forms (2) and (3) is usually made necessary by the design of the engine, which is such that the valve stem and eccentric cannot be placed so they will be in the same straight line. In the adjustment of such gears it is essential that the rocker be so located that it will vibrate equally on either side of a vertical line drawn through the fulcrum point. If the rocker is not adjusted as directed, the valve will receive a motion which may be faster in one direction than the other even though its travel is equalized. When such conditions exist it is impossible to secure a valve setting which will give a correct distribution.

Slipped Eccentric. A trouble which is sometimes experienced when the engine is in operation is caused by the eccentric becoming loose on the shaft and slipping around in such a way as to reduce the power of the engine or perhaps cause the engine to stall. This usually happens in engines where the eccentric is held

in position by means of a set screw. If a key is used the trouble is seldom experienced.

Method of Correction. When it does happen that the eccentric slips, the operating engineer can get a setting which, while not absolutely correct, will permit the engine to be operated with but little loss of time by following the directions here given:

First, set the engine on the head or crank end dead center, by inspection, with as much accuracy as is possible under the circumstances. *Second,* turn the eccentric around the shaft in the direction the engine is to run until steam will just begin to blow out at the cylinder drain cock on the end in question when the throttle is opened slightly. When this position is found, tighten the set screw in the eccentric temporarily. *Third,* turn the engine over to the other dead center and see if steam blows from the corresponding cylinder drain cock with the same degree of freedom. If it does the eccentric may be said to be in the correct position and the set screw may be securely tightened. When this is done the engine will be ready to again assume its duties. At the first opportunity the valve setting should be carefully checked by one of the methods described earlier in this text.

Increasing Power Capacity. It frequently happens in small plants using a plain slide valve engine that additional machines will be added from time to time until the engine finally becomes overloaded under ordinary conditions of operation. Under such circumstances the operator is asked to devise means of increasing the power delivered by the engine. This can be accomplished in one of the following ways: (1) by increasing the speed of the engine; (2) by increasing the pressure carried by the boiler; (3) by increasing the point of cut-off; and (4) by the combination of any two or all of the above methods.

Importance of Boiler Capacity. An examination into the methods given above reveals the fact that in every case additional load will be placed on the boiler. If the boiler capacity is sufficient to carry the additional load, then the problem can be solved, otherwise it cannot.

Increasing Speed. If the power is increased by increasing the speed of the engine to any very great degree, it will be necessary to change the size of the belt pulleys on the engine and line shaft in order not to disturb the speed of the machines.

Increasing Boiler Pressure. In increasing the power by increasing the boiler pressure, no changes are necessary unless it is thought advisable to replace any or all of the high-pressure steam pipe and fittings with extra heavy grade.

Lengthening Point of Cut-Off. If it is desired to increase the power by lengthening the point of cut-off, this can be accomplished by removing the valve and planing off the ends, thus reducing the steam lap the desired amount to give the increased cut-off. It is very essential to remove the same amount from each end of the valve, otherwise the steam lap would be different for each end. If the engine was originally cutting off at one-half stroke and it is desired to have the cut-off increased to three-fourths stroke, the amount of metal which should be removed to give the desired condition can easily be determined by drawing a Zeuner diagram from the valve in question. When the valve is finally reconstructed and placed in position in the steam or valve chest, it will be necessary to change the angle of advance of the eccentric in order to secure the proper amount of lead. To secure the proper setting it would be advisable to follow the directions given earlier in this text.

Use of Double Valve. As has been previously pointed out, the plain D-valve possesses certain objectionable features in the matter of steam distribution which is partially overcome by the use of a double valve. The Meyer valve is perhaps the most common form of double valve, a description of which is given on pages 73 to 78 of this text.

Setting Meyer Valve. In setting the Meyer valve, the main valve is set in the same manner as the ordinary simple D-valve. This main valve controls the admission, release, and compression points, while the riding, or secondary, valve controls the point of cut-off. Having correctly set the main valve, connect the riding valve to its eccentric and adjust the rods so that its travel is equal on each side of its central position, in exactly the same way as directed for the simple D-valve. When this is done, place the piston at the point where cut-off is desired and rotate the riding eccentric in the direction the engine is to run until a point is reached where the valve is just cutting off. When this point is reached fasten the riding eccentric to the shaft. Next place the

piston at the same relative position on the other stroke, and, if cut-off is just occurring, the valve may be said to be correctly set and the riding eccentric securely fastened. If cut-off does not occur at the same point on each end, make adjustments of the eccentric and valve rod until the cut-off points are equalized.

Pounding or Knocking. The question of pounding or knocking is discussed in "Steam Engines", but since this is frequently caused by improper valve setting, it seems well to give this troublesome matter a brief consideration.

Indications of Faulty Valve Action. If an annoying pound is heard which is difficult to locate, it is probably due to an improperly set valve. If this is the real cause of the trouble, it will be easily shown by indicator cards taken from the engine when under regular operating conditions. If the pound is due to valve action it will be revealed in the indicator cards in one or all of the following three things: (1) by compression beginning so early that the compression pressure exceeds the steam line pressure, thus causing the valve to be raised from its seat until the admission point is reached when the valve is forced to its seat with a "slam"; (2) by admission occurring so late that the lost motion first "runs out" and is then taken up after steam has been admitted; and (3) by the unequal distribution of power between the two ends of the cylinder, thus causing nearly all the work to be done in one end.

Correction of Fault. By following the directions as previously given for valve setting by measurement or by indicator, it becomes a comparatively small matter to correct the trouble.

CORLISS VALVE GEAR

Description. The Corliss valve gear is the most widely known of all the types of so-called "drop cut-off" valve gears. It is more economical than most other types from the standpoint of steam consumption but, on account of its peculiar construction and multiplicity of parts, is not adapted for high-speed work, say, above 100 revolutions per minute. Directions for setting a Corliss valve gear have been presented earlier in this text and need not be repeated here, but there is a word of caution which should be emphasized.

Possible Troubles. The rods connecting the steam valve arms with the dash pots should be adjusted so that when down

as far as they will go and with the wrist plate in its extremes of travel the stud die on the valve arm will just clear the shoulder on the hook die. If the rod is left too long, the steam valve stem will probably be bent, the valve arm broken, or the dash pot rod bent or broken. It may happen that the jar from the action of the dash pot will cause the dash pot rod to become loosened while in service. If this occurs the parts just mentioned may be broken in a manner similar to that when the rod is left too long in setting. Again, if the dash pot rod is left too short, the hook will not engage and, consequently, the valve will not open.

STEPHENSON VALVE GEAR

Extent of Use. The Stephenson gear, or *link motion,* as it is commonly called, is one of the oldest and best known types of reversing gears in use in the United States. For a great many years it was used almost to the exclusion of all other types of gears on American locomotives. Of recent years, however, its use has declined until today we find only comparatively few American locomotives equipped with the Stephenson reversing gear. The use of this gear is not confined entirely to locomotive service. In fact, it is made use of on steam engines in many classes of service, such as, steam tractors, steam road rollers, stationary engines, and hoisting engines.

Characteristics. *Increase of Lead in Open-Rod Construction.* One of the characteristics of the Stephenson reversing gear is that the lead of the valve increases from full to mid gear for open-rod construction and decreases from full to mid gear for crossed-rod construction. The crossed-rod construction is seldom used on engines unless service conditions are such as to make necessary its manipulation by the use of the reversing lever. The feature of increasing lead from full to mid gear, under certain conditions, is desirable on locomotives used for passenger service. In such instances the engineer will usually start the train with the reverse lever at or near the full gear position where the lead is a minimum and as the speed increases will bring the reverse lever nearer and nearer the central position where the lead is greater. This feature considered by itself is desirable since for best working conditions the lead should increase with the speed.

Back-Up Eccentric. One very desirable feature of the Stephenson gear is that it may be set to secure almost any steam distribution desirable. This is accomplished by making use of the "back-up" eccentric. Applying this method to setting the valves will, of course, disarrange the reverse, or "back-up", conditions but the "go-ahead" conditions can be almost perfectly secured.

Possible Troubles. *Lost Motion in Driving Boxes.* In the use of Stephenson gears on locomotives there is one condition which frequently arises but is rarely considered. The condition referred to is the development of lost motion in the driving boxes. In such cases, the eccentric being attached to the axle, the full amount of this lost motion is delivered to the valve with the link working in full gear. In certain other types of gears this condition would produce but very little change in the movement of the valves.

Effect of Vertical Motion of Engine. Another condition which affects the steam distribution when a Stephenson gear is used is the vertical motion of the engine on its springs caused by irregularities in the track.

Setting Valve. In setting the valve on an engine using the Stephenson gear, the fundamental principles involved are exactly the same as those given for the setting of a plain slide valve gear. We need to keep constantly in mind, however, that there are two eccentrics and two eccentric rods to deal with instead of one.

Typical Plain Slide Valve Setting. As an example let us consider the case of an engine fitted with a plain slide valve gear. Suppose it is desired to give the valve a lead of $\frac{1}{32}$ inch on both the head and crank ends. An examination of the valve discloses the fact that the lead on the head end is $\frac{1}{8}$ inch and that on the crank end is $\frac{1}{32}$ inch, which is the desired amount. The problem is to reduce the lead on the head end $\frac{3}{32}$ inch without disturbing the lead on the crank end. This problem can be solved by reducing the lead on the head end $\frac{3}{64}$ inch by changing the length of the valve rod and an additional $\frac{3}{64}$ inch by changing the angle of advance of the eccentric on the shaft. If the work is carefully done the results should show a lead of $\frac{1}{32}$ inch on both ends, unless the angularity of the eccentric rod is a very considerable amount.

Stephenson Valve Setting. Now suppose that the simple slide valve gear on this engine has been replaced by a Stephenson revers-

ing gear and that an examination of the valve with the reverse lever in full gear position shows the lead on the head end to be $\frac{1}{8}$ inch and that on the crank end $\frac{1}{32}$ inch for the forward position of the reverse lever, while for the backward position the leads on both the head and crank ends are found to be correct, namely, $\frac{1}{32}$ inch. In this case, the same as before, it is desired to secure a lead of $\frac{1}{32}$ inch on each end when the reverse lever is in both the forward and backward positions. To accomplish this with the reverse lever in the full forward position, it will be necessary to reduce the lead $\frac{3}{64}$ inch by changing the length of the eccentric rod and an additional $\frac{3}{64}$ inch by changing the position of the eccentric on the shaft. If the work is carefully done the desired results will be approximately secured.

Differences in the Two Settings. In the example just presented it should be noted that in the case of the simple gear the adjustments were made on the valve rod and eccentric, while in the case of the Stephenson gear they were made on the eccentric rod and eccentric. This correction of one-half the error on the eccentric and one-half on the eccentric rod, instead of on the valve rod, is necessary in order to permit the conditions on the reverse direction to remain unchanged. Other adjustments of a like nature can be made in a similar manner.

Variation in Conditions. Unfortunately, in practice, the Stephenson reversing gears are not always constructed so as to permit all the adjustments mentioned above. In such instances a compromise will have to be made.

WALSCHAERT GEAR

Extent of Use. The Walschaert gear has been used abroad for many years but never attained prominence in this country until ten or twelve years ago. It represents the most satisfactory type of radial reversing gear now in service in the United States. It is now being equipped on approximately 80 per cent of all new American locomotives, the remaining 20 per cent being fitted with the Stephenson gear. Its use, however, is confined almost exclusively to locomotive service.

Comparison with Stephenson Gear. One of the chief advantages of the Walschaert gear is the accessibility of all the parts and

the comparative ease with which repairs can be made. The parts of the gear being located outside, the space below the boiler may be used for other parts not so necessarily accessible. The chief point in which the Walschaert gear differs from the Stephenson gear on the action of the valve is that the former gives a constant lead for all positions of the reverse lever. Both gears are adaptable for use with any form of locomotive valve yet designed. The usual construction of the Walschaert gear is such as to permit little or no adjustments being made on the road. It is unusually free from any inclination of the parts to cause trouble through heating. Cases are known where improperly designed gears gave some trouble by the eccentric rod pins heating due to the twisting effect between the driving wheels and engine frame caused by unusual conditions of track and service. This, however, is a matter easily corrected.

Lost motion in the driving boxes produces much less effect on the motion of the valve when a Walschaert gear is used than when a Stephenson gear is employed. Neither does the up-and-down motion of the engine on its springs affect the steam distribution unless the connection of the eccentric rod to the link foot is placed at too high a point above the center line of the axle. In all well-designed Walschaert gears it is necessary that the trunnion upon which the link oscillates be fixed at an unvarying distance from the cylinder. In the fulfillment of this requirement it will be observed that the link bracket is invariably attached to the guide bearer, or yoke, and the slide for the valve stem is mounted on the upper guide bar. In some types of locomotives the construction is such that a large cast-steel bracket is laid across, joining the bars of the engine frame on both sides just back of the guide yoke, which acts as a frame binder and brace and a carrier for the link bracket. In still another type, the large casting is bolted to the guide yoke as well as the frame, thus forming a most substantial construction.

Repairs. With the Walschaert gear in service, if a break occurs within the valve gear, the difference in time consumed in making the temporary repairs necessary to get the engine moving under its own steam is greatly in its favor. This is one of the principal reasons for its adoption, since it means less time lost in delays.

EXAMINATION PAPER

VALVE GEARS

Read Carefully: Place your name and full address at the head of the paper. Any cheap, light paper like the sample previously sent you may be used. Do not crowd your work, but arrange it neatly and legibly. *Do not copy the answers from the Instruction Paper; use your own words, so that we may be sure you understand the subject.*

1. Name the various parts which go to make up the valve gear of a steam engine.

2. What is the function of the valve gear?

3. In the case of a plain slide valve, why is the valve seat usually constructed so that the valve travels over the edge of the seat a small distance?

4. What is meant by the *throw of an eccentric?* What will be the throw of an eccentric if the valve travel is $3\frac{1}{2}$ inches?

5. When is a plain slide valve said to be in mid-position?

6. What is meant by inside and outside lap, and how may they be determined in a given case?

7. What is *lead,* and why is it desired?

8. What effect does increasing the outside lap have on the steam distribution?

9. What effect does increasing the inside lap have on the steam distribution?

10. How do changes in the angular advance and eccentricity affect the steam distribution?

11. When speaking of the lap of valves, what is meant by *inside clearance* and when is it desirable?

12. What is meant by the *displacement* of the valve?

13. In the case of an engine having a plain slide valve, if the point of compression is different on each end, how may it be equalized? How would you equalize the point of cut-off?

14. What is the purpose of the Zeuner valve diagram?

15. State the distinction between a direct and indirect valve.

16. State briefly the method of procedure in setting a plain slide valve.

17. What is a balanced valve and what are its advantages?

18. What is the difference between the link and radial type of reversing gears?

19. In speaking of the Stephenson link motion, what is meant by "open" and "crossed" rods? What is meant by "hooking up"?

20. Name some of the advantages and disadvantages of the Stephenson link motion.

21. What is the principal difference between the Hackworth and Marshall reversing gears?

22. Name the advantages of the Walschaert valve gear over the Stephenson gear. What can you say in regard to the lead of the valve for the different positions of the reverse lever?

23. What are some of the advantages of double valves?

24. In setting the Meyer valve, is the main valve set to give an early or a late cut-off? Why?

25. What are some of the advantages of the Corliss valve gear? Can the Corliss gear be used on high speed engines? State your reasons.

After completing the work, add and sign the following statement:

I hereby certify that the above work is entirely my own.

(Signed)